RM E50A04

NACA RM E50A04

RESEARCH MEMORANDUM

APPLICATION OF BLADE COOLING TO GAS TURBINES

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
WASHINGTON

May 31, 1950
Declassified December 11, 1953

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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SUMMARY

A review of the status of knowledge on turbine-blade cooling and a description of pertinent NACA investigations are presented. Analyses of uncooled turbojet- and turbine-propeller-engine performance, including some relative range values, are used to illustrate the trends with changes in performance parameters and the gains to be obtained in some cases by operating at high gas temperatures. The current limitations in performance of uncooled and cooled engines are briefly discussed. Analyses of the performance possibilities of turbine-blade cooling are given for isolated blades, varying-design turbojet and turbine-propeller engines, and a typical jet engine. The possibilities when nonstrategic blade materials are used are included. Finally, the knowledge available and investigations to increase the knowledge on heat-transfer, cooling-flow, and performance characteristics of cooled turbines are discussed.

INTRODUCTION

The greatest efforts in aeronautical turbine research have been devoted to increasing the power that can be developed by an engine of a given size and weight, to increasing the power that can be realized from a given quantity of fuel, and to improving the reliability of turbines.

Although substantial increases in the power and the economy of turbines can be realized from improvements in the flow capacity and the efficiency (reference 1), the largest and most significant benefits may be derived from increasing the temperature of the working fluid at the turbine inlet and from increasing the compressor pressure ratio. For a given capacity or weight flow, it can be shown from turbine theory that the power per pound of air is proportional to the turbine-inlet temperature.

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*Based on lecture presented at Bureau of Ships, Department of the Navy, Washington D.C., May 10, 1949.

Current hydrocarbon fuels can produce far higher temperatures than are used now, but current turbine materials have insufficient strength at these higher temperatures to withstand the strains imposed. Although better heat-resistant materials are being developed, a more immediate means of realizing the performance improvement that should be possible with increased turbine-inlet temperatures is required. Another problem of great military importance is that, even if better heat-resistant materials are developed, both these materials and the high-temperature materials currently used contain a high percentage of scarce alloys. Unfortunately, metals with small quantities of these strategic alloys could not be directly substituted for the high-temperature materials because they have lower strength characteristics. Thus the problem exists of not only finding means to operate turbines at higher temperatures than currently used, but to accomplish such operation with low-alloy-content metals.

One method of attack on this problem is to cool the turbine, which has the possibility of attaining equal or lower metal temperatures with an increase in gas temperature. As a consequence of the potentialities of turbine cooling, the NACA started investigations on the cooling of turbines at the Lewis laboratory in 1945. Although other cooling problems are being investigated, only blade cooling will be discussed herein.

The NACA investigations prior to May 1949 are described and the following information is presented in detail:

(a) The gains to be obtained in performance through the use of high pressure ratio and temperature

(b) Detailed reasons for current performance limitations

(c) Reasons for turbine-blade cooling

(d) Some analytical comparisons of turbine-cooling methods

(e) Objectives of cooling research

(f) Some problems involved

(g) Status of knowledge on turbine-blade cooling

(h) Status and some results of investigations being conducted

ANALYSIS OF GAS-TURBINE-ENGINE PERFORMANCE

The theoretical performance gains possible through use of high temperatures at the turbine inlet and high compressor pressure ratios for turbojet and turbine-propeller engines are discussed. The turbojet engines were assumed to be made of materials that can withstand the stresses imposed by high pressures and temperatures, thus eliminating consideration of the cooling losses for these engines. In the consideration of cooling losses in turbine-propeller engines, the assumption is made that the turbine blades were water-cooled. Cooling losses are considered in one case and not in the other because these analyses are the only ones available. Nacelle drag is included in the calculations. The lift-drag ratio was such that the wing loading was limited to 80 pounds per square foot. The nacelle drag coefficient was 0.055 and the airplane drag coefficient was 0.0186. The structural weight of the airplane was assumed to be 0.4 of the gross weight and the tank weight, 10 percent of the fuel weight. The fuel weight was included. The details of other assumptions used in the engine-cycle analysis, which were similar to those usually used in such work, are not discussed herein because the analyses are not rigorous and only trends and not absolute values are shown.

Turbojet Engine

The computed variations of specific thrust and specific fuel consumption for a range of turbine-inlet temperatures and compressor pressure ratios for the turbojet engine are shown in figure 1. The calculations for figure 1(a) are based on an airspeed of 500 miles per hour, sea-level altitude, and compressor and turbine efficiencies of 0.90. The assumed compressor efficiency is somewhat high but such efficiencies are anticipated. The effect of efficiencies is small at the lower temperatures in any case. More detailed information on the effect of efficiencies on performance is presented in reference 1. Curves of relative range of the airplane are also plotted on figure 1, with the assumption of a base range of 1.0 for the current engine with a turbine-inlet temperature of 2000° R and a compressor pressure ratio of 4.

At any pressure ratio, the specific thrust and the fuel consumption increase with increase in turbine-inlet temperature. At a given turbine-inlet temperature, as compressor pressure ratio varies the specific thrust increases to an optimum value then decreases. The decrease is caused by the decrease in the permissible combustor temperature rise (as determined by compressor-outlet temperature) as the compressor pressure ratio increases. The optimum-power pressure ratio for each turbine-inlet temperature increases as turbine temperature is raised. Although the lowest values of specific fuel consumption occur at extreme pressure ratios, the specific fuel consumptions corresponding to the optimum-thrust pressure ratios are about 20 percent below the value for current temperatures and pressure ratios at the assumed component efficiencies and flight speed. For a given relative range, thrust increases appreciably and fuel consumption increases slightly as the turbine-inlet temperature is increased. The pressure ratio, of course, must be correspondingly increased. An indication of the gains can be obtained from a comparison of the engine at a turbine-inlet temperature of 2000° R and a compressor pressure ratio of 4 with the engine at a turbine-inlet temperature of 3000° R and a compressor pressure ratio of 16. Figure 1(a) shows that the specific thrust would increase from 52 to 93 (not quite double), the specific fuel consumption would decrease from 1.16 to 1.01 (about a 13-percent decrease), and the relative range would increase about 22 percent. These results mean that for a given required power the engine with the high pressure ratio and high inlet temperature would be much smaller and use less fuel than the current engine.

The effect of airspeed can be determined by comparing the results of figure 1(a) for 500 miles per hour with those of figure 1(b) for 1000 miles per hour. All other conditions are the same except a fixed lift-drag ratio of 7 was used at 1000 miles per hour. In general, the trends of the curves in figure 1(b) are the same as those in figure 1(a) except that the constant-temperature curves reach an optimum point at lower pressure ratios and the relative-range curves have more curvature than those in figure 1(a), causing specific fuel consumption to increase appreciably as turbine-inlet temperature is increased for any constant relative range. A combination of high thrust and low fuel consumption again requires a high turbine-inlet temperature and a high compressor pressure ratio. One trend that becomes evident in figure 1(b), which is true of all other figures of this type, is that, for a given turbine-inlet temperature, an optimum compressor pressure ratio exists when range is considered. This trend is shown by the results for a temperature of 2000° R. The greatest range is obtained at a compressor pressure ratio of approximately 10.

The effect of altitude on performance is obtained by comparing the data of figure 1(a) for sea-level altitude with those of figure 1(c), which is for an altitude of 30,000 feet; the other conditions are the same as those of figure 1(a). If the curves of figure 1(c) are superimposed on those of figure 1(a), the principal differences noted will be that for a given turbine-inlet temperature the compressor pressure ratio at which optimum specific thrust occurs increases as altitude increases and that specific thrust for a given temperature and compressor pressure ratio is greater at high altitudes than it is at low altitudes. A virtual lifting of the whole diagram occurs with a shift to the left. The greater specific thrust at altitude for a given inlet temperature and pressure ratio is caused by the ratio of inlet temperature to ambient temperature being greater than at sea level. In general, the same trends occur as at low altitude, appreciable gains in specific thrust and decreases in specific fuel consumption occurring if gas temperature and pressure ratio are greater than current values.

In order to illustrate the significance of figure 1(a), a comparison of the current engine with the potentialities of the optimum power combination is considered, temporarily neglecting the secondary variation of specific cycle performance with altitude. If a sea-level specific mass flow of 15 pounds per second per square foot of frontal area and a gross thrust of 6000 pounds are assumed, the engine with a turbine-inlet temperature of 2000° R operating at a pressure ratio of 4 would require a mass flow of 114 pounds per second, which corresponds to a frontal area of 7.6 square feet, and would have a specific fuel consumption of about 1.16 pounds per pound thrust. An equivalent engine with a turbine-inlet temperature of 3000° R operating at optimum-thrust pressure ratio of 16 would require a mass flow of only 65 pounds per second corresponding to a frontal area of about 4.3 square feet and would have a specific fuel consumption of about 1.01 pound per pound thrust. If, in order to extend the range of the aircraft, the thrust were reduced to 4250 pounds by means of reduced turbine-inlet temperature at constant initial pressure ratio, the relative range of the high-pressure-ratio engine would be increased to about 1.4 at a fuel consumption of about 0.83 pound per pound of thrust, and that of the low pressure-ratio engine to 1.10 at a fuel consumption of 1.00 pounds per pound of thrust.

The high-performance engine with the small frontal area is seen to have a substantial advantage in that for a given maximum thrust capacity, its range at reduced cruising conditions is approximately 18 percent greater than the low pressure-ratio engine. Conversely, if the two engines were of the same frontal area and had the same reduced or cruising thrust for extended range, the maximum thrust theoretically obtainable for take-off and combat with the high-performance engine would be about 70 percent greater than that of the conventional low-pressure-ratio engine.

The disadvantage of an engine with a given compressor pressure ratio having its design point at a high gas temperature is that the range is penalized and economical reduction of power for cruising is impossible. The disadvantage of an engine with the same pressure ratio having its design point at a turbine-inlet temperature of 2000° R is that no margin of excess power is available for take-off, climb, and combat maneuvers. The addition of thrust augmentation to the low-temperature long-range engine through the use of a tail-pipe burner penalizes weight and cruising performance. For subsonic flight speed, it is evident from figure 1(a) that an engine design having operating points available at both 2000° and 3000° R would be highly desirable because, for the given compressor, maximum range can be obtained at cruising conditions with an adequate margin of high thrust for other operations when required.

When cooling research has progressed to a point such that cooled turbines can be designed with confidence, the gas temperature and the type of engine installation will depend on the type of aircraft as well as on results of many more analyses of the type presented, which must be made to cover a broader range of conditions.

Turbine-Propeller Engine

Some typical performance results of a turbine-propeller engine, are shown in figure 2. Curves of constant relative range are again shown. The great difference between these results and those for the turbojet engine is that, for constant pressure ratio and increasing turbine-inlet temperature, the specific fuel consumption decreases, whereas the power greatly increases. Another noticeable feature of these curves is that the optimum compressor pressure ratio, at a given temperature at which the relative range is a maximum, is very marked. If the engine with a turbine-inlet temperature of 2000° R at a compressor pressure ratio of 4 is considered as the current engine, increasing the temperature to 3000° R and the pressure ratio to 16 will result in an increase in power from about 105 to 300 horsepower per pound of air per second (a threefold increase), a decrease in specific fuel consumption of about 40 percent, and an increase in range of about 70 percent. These results are remarkable and warrant great effort to attain them in actual practice.

The effects of altitude and flight speed are obtained by comparing the results of figure 2(a) with those of figure 2(b) at a flight speed of 400 miles per hour and an altitude of 30,000 feet.

The same remarkable improvements shown in figure 2(a) can be obtained for the conditions stated for figure 2(b) when compressor pressure ratio and turbine-inlet temperature are increased above current values. The lower constant-relative-range and compressor-pressure-ratio curves show the same trends as those for the turbojet engine in that as temperature increases the specific fuel consumption increases.

The potential improvements resulting from the use of high-cycle temperatures and compressor pressure ratios in the turbine-propeller engine are of great interest, particularly with respect to long-range aircraft. The analyses given are only approximate because the cooling evaluation has not been sufficiently explored for complete evaluation of the problems.

CURRENT LIMITATIONS ON PERFORMANCE

The limitations on the performance of engines using both uncooled and cooled turbines will be discussed in this section of the report. The limits imposed on the uncooled turbine engine are compressor-pressure-ratio limits and material-temperature limits. The limits to the cooled turbine engine include cooling losses and design complications in addition to those mentioned for the uncooled turbine engine.

Uncooled Turbines

Compressor pressure ratio. - The compressor pressure ratios of current engines range from about 3.5 to 5.5. Much research is being conducted to increase this ratio in both centrifugal- and axial-flow compressors. Developments in subsonic axial-flow-compressor design are expected to yield pressure ratios of 1.34 per stage, which for a pressure ratio of 27 would require 12 stages as compared with 25 stages at current stage pressure ratios. Supersonic axial-flow stages and subsonic mixed-flow stages with ratios up to 4 are considered feasible and the pressure ratio of 27 could be accommodated with a two-stage supersonic axial stage followed by a mixed-flow stage having a pressure ratio of 1.7. From the research that has been conducted, the problems of attaining high pressure ratios apparently are not insurmountable although mechanically difficult.

Materials. - Uncooled gas-turbine-engine performance is limited because the temperature at the turbine inlet cannot exceed about 2000° R. This limitation is caused by the insufficient strength of currently available turbine materials at temperatures higher than 2000° R. Existing hydrocarbon fuels can produce, however, far higher temperatures, approximately 4000° R.

A problem of grave strategic importance is the high percentage of very scarce costly alloys in these materials and, in addition, some of the alloys must be imported from other countries. In the event of war, grave doubt exists that enough of these materials could be supplied to manufacture the number of engines required by a greatly expanded air force. As a consequence, substitute metals would have to be used, resulting in an uncooled turbine of a lower specific thrust than is currently obtained because the operating gas temperature would have to be decreased and consequently a larger engine would be necessary to keep the thrust constant at its present value.

The most critical elements in order of importance are: cobalt, columbium, chromium, nickel, and tungsten. If materials containing cobalt and not more than 0.5-percent columbium are eliminated, most of the familiar high-temperature materials would be unavailable. These materials are N-155, K42B, S-590, S-816, Vitallium, X-40, Inconel X, and many others. The high-temperature alloys then remaining are 19-9 W-Mo, 19-9 DL, 16-25-6, Hastelloy B, Inconel, Discaloy, Nimonic 80, and 17W. With the exception of Hastelloy B, these materials all require from 10- to 20-percent chromium, which is third on the critical list, and up to 75-percent nickel, which is fourth. If it is further specified that the alloy contain not more than 40-percent chromium plus nickel, Nimonic 80, Inconel, and Hastelloy B are eliminated and the following materials remain: 16-25-6, 19-9 DL, 19-9 W-Mo, Discaloy, and 17W. These materials constitute a class of relatively nonstrategic alloys, which could probably be broadened with further development. Chromium content can be further reduced with addition of silicon and titanium in small quantities and nickel can be reduced with addition of molybdenum, which is lower on the critical list. Typical of these reduced materials are the steam-turbine alloys, such as Sichrome No. 1, with 8.5-percent chromium, 3-percent silicon, 0.6-percent manganese, and no nickel.

Only materials having not more than 4- or 5-percent total strategic-alloy content should practically be considered. The most important problem is therefore to find some means of using such materials without effecting a decrease in turbine-inlet temperature below its present value or preferably with an increase in this temperature above its present value to obtain the specific power increases desired.

Cooled Turbines

Cooling losses. - The immediate means for solution of the foregoing problem is engine cooling, especially cooling of the turbine blades. The apparent simplicity of putting air or water through blades, however, is misleading in many respects. Many phenomena arise that limit the performance. Although cooling at current turbine temperatures does not seem to offer serious difficulties, the study of high-performance cooled engines involves many additional complications, which cannot be generalized but are functions of a particular engine-design configuration. The performance of a cooled engine will inevitably be inferior to a similar uncooled engine, and the turbine-cooling requirements are dependent on many variables associated with the purpose to which the power plant is put, operating conditions, and cycle characteristics. The problem therefore is one of economics of the entire propulsion system, advantages of cooling as related to use of noncritical metals, energy balances, and net gains in performance.

The losses associated with cooling can readily become excessive both in current engines and the high-performance engines contemplated for the future, which will result in limited engine performance. These losses include heat loss to the coolant when the coolant does not mix with the combustion gases, losses due to pumping of the coolant through the system including the blades, power loss due to the compression of air (if it is the coolant) in the engine compressor, drag of a radiator in the case of liquid coolants, loss in working fluid for the turbine as a result of bleedoff from the compressor for coolant purposes, and a rather intangible loss in blade power because the working fluid has heat abstracted even though this heat may be regained by cooling air mixing with the combustion gases at the blade tips. Keeping these losses to a minimum so that the lowest possible limitation in performance is obtained requires highly refined techniques in analysis and design as well as reliable and comprehensive data.

Materials. - The limitations in performance arising because of scarcity of certain alloys applies, of course, to cooled as well as to uncooled turbines; that is, cooling must be applied to turbines, if at all possible, where blades are made of low-alloy-content materials. The performance of an uncooled turbine will naturally be less than that of a cooled turbine with high-temperature-material blades.

In cooled blades, thermal stresses, which are not normally present in uncooled blades, will inevitably arise and may limit the performance. The extent of limitation will probably depend on the material used. Vibratory stress is also expected to be a very difficult problem in air-cooled blades because the blade is a thin-wall structure and a large number of complex modes will be introduced.

Design complications. - The specific power of a gas-turbine engine (which is the power/(lb air) (sec)) is appreciably increased as the turbine-inlet temperature is increased, but for a turbine of a given design, the flow (lb/sec) decreases as the turbine-inlet temperature is increased. Both turbine speed and compressor pressure ratio must therefore be increased in order to retain constant flow capacity and flow similarity with fixed turbine-nozzle area and angle; this increase results in increased turbine-rotor and blade stress in order to realize the potential improvements in specific power of the turbine in the turbine-propeller engine. Because of the added stress, the performance of the turbine-propeller engine may be limited to some value below that theoretically obtainable from the increased turbine-inlet temperature. In order to overcome this limitation, the high-temperature turbine should be a new design with altered geometry and new blade configurations so that the rotor speed can remain the same as in the low-temperature original design. Thus, complications arise in trying to apply high temperatures to a gas-turbine engine in that a practically new design is required for some parts.

One problem that has great possibility of limiting performance of the air-cooled turbine is that of choking in the cooling-air passages of the blades; that is, the Mach number will reach a value of 1 in the blade passage under some conditions and no more air can pass through the blade even though it may be required. Design changes are thus required to enlarge the blade passages; therefore the blade has to be thicker and fewer blades can be used for a given diameter. It is also evident from preliminary calculations that the blade taper from root to tip must be as small as possible to reduce acceleration of the air through the passage. This acceleration is tremendously rapid because of centrifugal action and heat absorption by the cooling air. The entire turbine may therefore have to be redesigned so that the performance may not be limited by the choking phenomena. This phenomena is greatly accentuated as altitude increases.

A design complication, which causes losses and thus limits the performance of a cooled engine, is the bleedoff of air from the compressor for cooling. The system conveying the air from the compressor to the turbine must be added and has inevitable pressure losses.

In a high compressor-pressure-ratio engine, the cooling air must come from one of the last stages of the compressor in order that it will have sufficient pressure at the blade tip to cause positive flow from the blade passage into the surrounding gases, which are at a high pressure. It is very possible that this air, because of the compression, will have a temperature at the blade root, which will exceed an allowable limit. The reduction of this temperature in any case has a very beneficial effect on cooling. Thus the complication of an aftercooler between the compressor and the turbine is added to the engine with an adverse effect on performance due to weight and drag. Because of the very beneficial effect that reducing the cooling-air temperature at the blade root has on reduction of cooling-air requirements, however, the addition of an aftercooler may have a beneficial effect on over-all performance rather than becoming a device that adds to the losses and thus limits performance.

In the case of liquid cooling, a radiator and a piping system must be added. Additions of weight because of these added parts and the presence of the coolant in the aircraft plus the pressure losses in the system and the possible drag of the radiator will reduce the performance of the engine. Because of heating of the air passing through the small radiator, which cools the turbine-blade liquid coolant, a small radiator thrust instead of drag may occur, as noted in reference 2.

The use of high pressure ratios will increase the stresses due to gas forces, which will require strengthening of the engine parts. The saving in fuel weight, however, occasioned by high pressures plus the gains in specific power occasioned by corresponding increases in turbine-inlet temperature, which may be expected with cooling, and for a given power requirement will result in smaller engines, are expected to more than offset the added engine weight occasioned by higher stresses. If materials are used such that the temperature cannot be increased, a possibility of added weight exists if pressure ratio alone is increased.

Many other factors add design complications and may limit performance of cooled turbine engines. Only a few factors are presented to show that the cooled turbine engine may not be a simple device and that much thought is required in its design.

PERFORMANCE POSSIBILITIES OF TURBINE-BLADE COOLING

The performance possibilities of turbine-blade cooling are discussed in three parts: (1) Several methods of cooling turbine blades are compared; (2) the use of nonstrategic blade metals (Cr-Mo-Va and SAE 1015) are compared with a strategic blade metal (S-816); and (3) the cooling-air requirements of both a turbojet and turbine-propeller engine are evaluated for several operating conditions.

Relative Effectiveness of Cooled Blades

Analysis method. - The performance possibilities of turbine-blade cooling have been determined using the allowable effective gas temperature as the criterion for the effectiveness of various methods of cooling blades of different configurations and of various materials. Many methods have been suggested for accomplishing the desired cooling of turbine blades. The decision as to which of these methods or what combination of these methods has the best practical value can be made only after a large amount of experimental data has been obtained and studied. The mechanism of flow in boundary layers surrounding turbine blades is not yet sufficiently understood to make accurate quantitative comparisons by analysis. If the cooling of the boundary layer is neglected, however, much can be learned about the suitability of various forms of conduction cooling by using the principles originally set forth by Fourier. Suitable average values may be assigned to the heat-transfer coefficients between the hot gases and the metal and between the metal and the cooling fluid. Allowances may also be made for radiation.

In order to evaluate the temperature distribution in a cooled turbine blade, the following parameters must be specified: the turbine structures, the shape and the material of the blades, and the temperatures, the rates of flow, and the composition of the hot gases and the cooling fluid. The convection and radiation coefficients can then be determined if the laws are known. The usual equations for radiation were used to evaluate these effects with a geometry factor of 1.0 being used. The equation for fully developed turbulent flow inside pipes has been used to determine an average convection coefficient from metal to coolant. The equations for radiation and inside convection coefficients were obtained from reference 3. The temperature of the gas effecting heat transfer (called the effective gas temperature) $T_{g,e}$ was assumed to be constant around the blade and was determined from the equation

$$1 - \Lambda = \frac{T_g'' - T_{g,e}}{T_g'' - T_g} \tag{1}$$

where the recovery factor Λ is roughly equal to the square root of the Prandtl number $(c_{p,g}\mu_g/k_g)$ of the gas (reference 4). (All symbols are defined in the appendix.) The static temperature of the gas T_g is determined from the turbine-inlet total pressure and temperature, the gas flow, and the turbine-nozzle area for a Mach number of 1.0 or less at the rotor inlet. When the pressure ratio is such that expansion occurs between the nozzle outlet and rotor inlet causing the Mach number to be greater than 1.0 at the rotor inlet, the static gas temperature calculated by the method mentioned would be unsatisfactory. Such a case was not encountered in these calculations. The total temperature of the gas relative to the moving rotor blades T_g'' is determined from the static temperature and the relative velocity of the gas to the blade W_g using the equation

$$T_g'' - T_g = W_g^2/2Jgc_{p,g} \tag{2}$$

The effective gas temperature $T_{g,e}$ is required because the convection heat-transfer coefficient on the gas side is based on the difference between the effective gas and blade temperatures.

From the foregoing data, the radial-temperature distribution for any cooling arrangement and at any gas temperature can be found by formulating the equations expressing the heat balance at each infinitesimal part of the turbine blade and performing the required mathematics. A curve like the solid one presented in figure 3 showing the blade temperature from the root to the tip will be obtained. A variation in effective gas temperature will cause the curve to move up or down.

In order to determine the amount by which the temperature of the hot gases can be increased by a given method of cooling, the temperature distribution shown by the solid line in figure 3 must be compared with the allowable temperature distribution. This temperature distribution is obtained from the centrifugal stress in the blades and stress-rupture data because of lack of knowledge concerning other stresses involved. Thermal stresses are involved and a relation is needed between thermal, centrifugal, and gas-bending stresses for each particular blade configuration. A turbine blade in operation is also characterized by many complex modes of various orders of the exciting frequencies induced by combustion chambers, nozzles, or aerodynamic flutter. Because of the many frequencies of higher order and the frequency variation from blade to blade, avoidance of all critical frequencies in each blade all the time is very difficult. Because the higher modes of vibration are of high frequency, the criterions of blade failure due to vibration should be the endurance limit for infinite life because any finite number of cycles would be consumed in a short period of turbine operation. Adequate data on elevated-temperature endurance limits are unavailable. Much experimental work remains to be done on the vibratory stresses.

Before the stress-rupture data are used, an allowable life is assumed and the relation between the maximum allowable temperature of the blade material and the blade stress is established as follows: When the tip speed of the blade is specified, the radial stress distribution of the blade is determined and the maximum permissible temperature at each radius of the blade may then be computed. A curve, such as shown by the dashed line in figure 3, may then be drawn to show the radial distribution of the stress-limited temperature.

The maximum effective gas temperature for a given tip speed is established when the actual temperature-distribution curve (solid line, fig. 3) has a point of tangency (circle, fig. 3) with the stress-limited or allowable blade-temperature-distribution curve (dashed line, fig. 3) for the same speed. When the curves intersect, the sections on the blade within the overlapping regions are too highly stressed. The effectiveness of each cooling method, material, and blade-configuration combination is presented relative to that for the uncooled solid turbine blade made of one of the most frequently used high-temperature materials, S-816. The relative effectiveness presented hereinafter is the ratio of the maximum allowable effective gas temperature in $^{\circ}R$ for the cooled blade to the maximum allowable effective gas temperature in $^{\circ}R$ for the uncooled blade. Such a relative effectiveness was used because in these simple analyses the heat-transfer coefficients for the air- and water-cooled blades were held constant and thus the results are only comparative. The effects of altitude on coefficients, losses, and so forth do not appear.

This procedure has been applied to three methods of cooling:

(1) Rim cooling, in which the temperature of the rim of the turbine disk is reduced by some cooling method and the solid turbine-rotor blades are cooled by conduction of heat to the rim

(2) Cooling of blades by passing air through passages of various shapes inside the blades

(3) Cooling by liquid, which circulates through passages formed in the blades and disks; heat flows through the metal to the cooling fluid

Comparison of cooling methods and blade configurations. - The effectiveness of rim cooling for a wide range of cooling variables and blade shapes is discussed in reference 5 and the results are summarized in figure 4. The relative effectiveness is plotted against the difference between the effective gas temperature and the temperature at the blade root for each of a series of cooling parameters. The dimensionless parameter includes gas-to-metal heat-transfer coefficient H_o, the perimeter of the blade section l, the blade length L_B, the thermal conductivity of the blade material k_B, and the cross-sectional area of the blade section A_B. For modern gas turbines of high power and heat-transfer coefficient, the perimeter and the blade length are large, whereas the thermal conductivity is small. For example, the value of the parameter would be 14 for turbines similar to those of current jet engines. Under these circumstances, cooling is indicated by the lower curves, and relative effectiveness is only about 1.05.

If metals with the conductivity of copper and the strength of current steels were available, the value of the parameter would be about 1.47 and the effectiveness would be appreciable.

Although this increase in effectiveness for current jet engines is not nearly enough to permit the use of very high gas temperatures, it is enough to cause a significant gain in estimated turbine-blade life.

The effectiveness of a more direct method of cooling, that of passing air through passages inside blades, is shown by results plotted in figure 5. The equation used to obtain the blade-temperature distribution is based on detailed derivations given in reference 6. The relative effectiveness is plotted against the ratio of weight of cooling air flow to weight of gas flow, called the dilution, for several methods of air cooling and several inside blade configurations. The material used in the blades is S-816. The lower curve is for a plain thin-walled hollow blade with cooling air flowing through the blade from the root to the tip. With a ratio of coolant flow to gas flow of 0.05, the relative effectiveness is about 1.25.

The placing of an insert in a hollow blade, as shown by the sketch in figure 5, forces the cooling air to flow between the insert and the blade wall. This modification increases the air velocity through the passage, as compared with that for the hollow blade, and increases the cooling effectiveness above that for a hollow blade, as shown by the results in figure 5. For a ratio of coolant flow to gas flow of 0.05 or 5 percent, the relative effectiveness for the blade with insert is 1.65 as compared with 1.25 for the hollow blade.

The cutting of slots in the walls of a hollow blade, as shown by the sketch of a film-cooled blade in figure 5, allows cooling air to flow out of the slots and over the blade surface, forming a cooling insulating film of air between the hot gas and the blade wall. This method of cooling is called film cooling. From data available in reference 7, the film-cooling curve in figure 5 was obtained. The relative effectiveness of the film-cooled blade is very good, being about 1.8 at a ratio of coolant flow to gas flow of 0.05 as compared with 1.65 for the blade with the insert. The cooling air issuing from film-cooled blades may upset the flow of gases about the blade. Because the air issues nearly tangentially to the gas flow, it is expected that the effect will be small.

Inasmuch as the results of the film-cooled blade are so favorable, putting a large number of holes in the blade surface seems warranted. This condition exists when the blades are made of porous material and cooling air is allowed to seep through the thousands of microscopic holes in the material from the inside surface to the outside surface of the blade wall. The results of such a method of cooling, calculated from data in reference 8, are shown in figure 5. The relative effectiveness is much greater than that for the film-cooled blades, about 3.5 at a ratio of coolant flow to gas flow of only 0.03. These calculated results in figure 5 are questionable in that the data were obtained on round specimens. The data were applied to the blade shape by using the areas of the blade involved.

Porous cooling is considered adaptable to turbine stator blades but for the rotor blades doubt exists as to whether porous material will have adequate strength characteristics. Because of this doubt, another blade configuration was studied. This configuration has fins along the entire passage length, as shown by the sketches of the 12- and 25-fin blades in figure 5. The fins are

0.020 inch thick and are spaced 0.040 inch apart in the 25-fin blade; they are 0.040 inch thick and are spaced 0.085 inch apart in the 12-fin blade. The addition of fins greatly increases the surface in contact with the cooling air and has been of infinite benefit in cooling internal-combustion engines for aircraft use. The results in figure 5 show that the fins are very effective; the relative effectiveness for the 25-fin configuration is about 2.4 at a ratio of coolant flow to gas flow of 0.05 as compared with 1.25 for the hollow blade. From these results, it is thought that blade configurations are available that will make possible the attainment of temperatures of at least 3000° R at the turbine inlet with air cooling using high-temperature or porous materials.

The effectiveness of liquid-cooling using either water or kerosene as the coolant is shown in figure 6. The analysis assumes that coolant passages are close to critical areas. Again the material of the blades is S-816. The curves are based on the 1000-hour stress-rupture properties of the metal. Details of the equations used to obtain blade-temperature distributions when the coolant passes through holes in the blade, as shown in figure 6, are given in references 9 to 11. Round holes must be used because of the high water pressures obtained at the high rotative speeds of the turbine rotor. The curve for water illustrates the remarkable theoretical effectiveness of this coolant. For a ratio of coolant flow to gas flow of only 0.02, the relative effectiveness is about 2.2. This effectiveness results from the apparent little difficulty in obtaining high inside heat-transfer coefficients with water.

Consideration of liquid cooling leads to the possibilities of the use of fuel as a coolant. This use would minimize the heat loss because the heat energy would reappear in the combustion chamber when the fuel is burned. The effectiveness of kerosene is much lower than water, however, because the heat-transfer coefficients are only about one-fourth the values obtained with water. The curve in figure 6 labeled fuel-air ratio is the ratio of the fuel required to obtain the gas temperatures corresponding to the relative-effectiveness values on the ordinate to the gas flow passing through the turbine. Fuel-air ratio and ratio of coolant flow to gas flow have the same value along this line. At a ratio of coolant flow to gas flow of approximately 0.03, the curve of fuel-air ratio required for combustion crosses the curve of relative effectiveness obtained with kerosene. At this point, all the fuel used for engine operation is needed for cooling. The value of relative effectiveness is about 1.25. If a higher effectiveness is required, more kerosene than is needed for combustion would be required for cooling; thus recirculation of some of the fuel and, consequently, a heat exchanger would be required.

Usually an effectiveness greater than 1.25 will probably be required. Consequently, a heat exchanger would generally be necessary for the blade configuration considered even though fuel is used as a coolant.

Comparison of various metals. - The relative effectiveness of the hollow air-cooled blades, the air-cooled 25-fin blade, and the liquid-cooled blade when they were made of various metals was next determined. The 25-fin blade was considered because it was so superior to the other configurations considered applicable to rotor blades. The hollow air-cooled blade was considered in order to determine whether low-alloy-content materials could be used in a blade that is easily fabricated and at the same time the turbine could be operated at current gas temperatures. In all cases, results were obtained using blades made of S-816, Cr-Mo-Va, and SAE 1015. The alloy S-816 has about 95 percent of alloying metals, many of which are scarce; Cr-Mo-Va has only about 5 percent of strategic alloying metals and about 95-percent iron, and SAE 1015 is ordinary automobile fender steel with no strategic alloying metals. If SAE 1015 is used, a coating must be applied to protect it from oxidation and corrosion by the hot gases. About 2 percent of alloying metal is required to prevent oxidation and corrosion.

The results for the hollow blade in figure 7 indicate that with Cr-Mo-Va blades, a ratio of coolant flow to gas flow of 0.05 is required to obtain a relative effectiveness of 1.0, or to operate at current temperatures. The curve for SAE 1015 never reaches a value of relative effectiveness of 1.0. It is evident that something more efficient than the hollow air-cooled blade is required if current gas temperature or slightly greater temperatures, which are necessary to overcome cooling losses, are to be obtained with nonstrategic materials without using great quantities of cooling air. This requirement leads to the 25-fin blade results, which are shown in figure 8. With a ratio of coolant flow to gas flow of 0.05, the relative effectiveness of this blade made of Cr-Mo-Va is 1.75 and even when made of SAE 1015 it is about 1.4 at the same ratio. Thus, such a configuration with air-cooling provides a solution to the double objective to be obtained by operation at higher gas temperatures and use of nonstrategic materials.

The effectiveness of liquid cooling when the blades are made of various metals is shown in figure 9. As in the air-cooled 25-fin blade, the water-cooled blade has a high relative effectiveness even when nonstrategic materials are used. The results with kerosene cooling are much lower but a relative effectiveness greater than 1.0 can be obtained at high coolant flows with Cr-Mo-Va blades. A relative effectiveness of more than 1.0 is impossible with SAE 1015 blades. The S-816 blades have a relative effectiveness of 1.8 at a ratio of coolant flow to gas flow of 0.10, which is required for a relative effectiveness of 1.0 for the SAE 1015 blades. Calculations were made to determine the effectiveness of water-cooling using 25S (aluminum) blades. The results in figure 9 show that even with 25S at a ratio of coolant flow to gas flow of 0.10 the relative effectiveness is greater than 1.0. Both the high conductivity and the light weight of 25S are favorable factors in obtaining high gas temperatures. It has low-strength characteristics, however, which is an unfavorable factor.

The analytical studies of liquid cooling showed that the cooling passage must be placed close (about 1/4 in.) to the trailing edge (reference 10) in order to have satisfactory trailing-edge temperatures. The trailing-edge temperature increases rapidly according to current analytical methods, which neglect cooling effects of the gas temperature in the boundary layer, as this distance is increased.

Cooling Requirements for Engines

Because of the promising results of the studies on possibilities of turbine-blade cooling using simple analyses, further studies were made of the cooling requirements for engines using more rigorous analyses. Studies were made of a varying-design turbojet engine, a varying-design turbine-propeller engine, and a typical turbojet engine.

Varying-design turbojet engine. - The cooling-air requirements have been determined for a series of turbojet engines as a function of blade-metal properties and altitude for a range of turbine-inlet temperatures at optimum-power pressure ratio thus defining the principal factors affecting the cooling process. A 12-fin blade configuration was used. The turbine operating conditions were determined from a relatively simple cycle analysis for a range of altitudes; the turbine staging was selected to be consistent with an average blade relative Mach number of 0.8 for the profile; the heat-transfer coefficients for the profile were computed from unpublished correlations obtained in a static cascade of blades; the equation for one-dimensional heat transfer (including the effects of centrifugal compression of the air in the cooling passage, reference 6) was used to determine the blade-temperature distribution; and the temperature distributions so obtained were matched with the allowable blade-temperature distribution as fixed by tip speed, centrifugal-stress distribution, and the 1000-hour stress-rupture characteristics of the materials in question.

The results of this analysis for the first stage of the turbine are given in figure 10. Ratio of coolant flow to gas flow is plotted against allowable effective gas temperature for two altitudes and for two materials, S-816 and Cr-Mo-Va, the nonstrategic material. The compressor pressure ratio of each engine was optimum from a power consideration for the temperature used. The assumed compressor and turbine efficiencies were 0.85. It is immediately apparent that, with the assumed conditions, the required ratios of coolant flow to gas flow rapidly increase with simultaneous increase in turbine-inlet or effective gas temperature and compressor pressure ratio. This rapid increase results from two principal variations. Obviously, the temperature difference between the gas and the blade increases; but, in addition, the blade heat-transfer coefficient is independently a function of local static pressure within the stage because the heat-transfer correlation is based on boundary-layer conditions. Compressor pressure ratio, turbine-stage pressure ratio, flight speed, and altitude thus appear directly in the density term of the Reynolds number, whereas in the conventional correlation the Reynolds number is a function of the mass velocity of the stream, and the heat-transfer coefficient would be independent of pressure ratio at constant mass velocity. Increasing altitude for a given engine (temperature and pressure ratio fixed) results in substantial increases in desired ratios of coolant flow to gas flow. An explanation for the increase in required ratio of coolant flow to gas flow with altitude can be seen from the equation derived for blade-metal temperature (reference 6) in which metal temperature is a function of the ratio of outside and inside heat-transfer coefficients, the outside heat-transfer coefficient, and the coolant flow. The effect of fins in the passage is to magnify the effect of coolant flow on the inside heat-transfer coefficient. The reduction of density with altitude therefore has a greater effect on the inside heat-transfer coefficient causing the increase in requirements of ratio of coolant flow to gas flow with altitude. For a given set of conditions, the ratio required is also appreciably increased when the nonstrategic material is substituted for S-816. At high altitudes the cooling-air Mach numbers for a given ratio of coolant flow to gas flow are appreciably higher than those at sea level. The ratio of coolant flow to gas flow that can be used depends on the losses obtained, which have to be included in an over-all analysis.

The results of the analysis are only approximate for the following reasons: The variation of blade height with cycle design conditions was not included, which makes the results pessimistic; in practice the blade height would be decreased at the high temperatures and pressure ratios, and lower ratios of coolant flow to gas flow than those indicated in the curves would be necessary; the limits of permissible ratio of coolant flow to gas flow as determined by losses have not yet been established for the turbojet engine but there is reason to believe that they are higher than those permissible in the turbine-propeller engine.

The conclusions that can be drawn currently are that for the high-performance turbojet engine, when the limitations of the analysis are considered, the application of nonstrategic materials is feasible for turbine-inlet temperatures up to 3000° R (effective gas temperature of about 2750° R) at optimum power pressure ratio. Nonstrategic materials also must develop the highest possible physical properties through careful selection and heat treatment; the blade profiles must be designed to favor minimum surface heat-transfer coefficients; and the internal-passage configuration must be highly refined to permit maximum effectiveness. The application of nonstrategic materials to high-performance engines presents formidable problems requiring extensive research, but the gains in performance and the strategic necessity for reduction in alloy content justify the effort. The potentialities at current turbine temperature and pressure ratios probably can be readily attained with development of the finned-blade configuration.

Varying-design turbine-propeller engine. - Application of air cooling to the turbine-propeller engine has been analyzed to some extent for the optimum-power pressure-ratio conditions at an altitude of 35,000 feet using a 25-fin blade, similar to that shown in figure 5, made of S-816 alloy. The required first-stage ratio of coolant flow to gas flow was found to increase rapidly from about 0.02 at an effective gas temperature (approximately 0.90 times turbine-inlet temperature) of about 2100° R to 0.18 at about 3100° R, as shown in figure 11. The effect of simultaneous increase in gas temperature and cycle pressure ratio is similar to that in the turbojet analysis. The results are pessimistic for the same reasons given in the turbojet analysis.

Additional calculations of the flow in the internal passage made using current compressor pressure ratios including the simultaneous effects of friction loss, momentum change, and centrifugal compression indicate that the passage would choke near the blade tip for the ratio of coolant flow to gas flow of 0.18 and a positive pressure ratio of about 2 from blade root to tip was required in addition to the centrifugal compression in order to overcome the friction and momentum-pressure losses. With a pressure ratio of slightly over 3 per stage, which was assumed, calculations showed that for the pressure-ratio values given the cooling air would have to be bled from the compressor near the discharge end at a pressure ratio of approximately 25. Because of the rapid state changes in the turbine, the second-stage ratio of coolant flow to gas flow was only 0.02 at an effective gas temperature of 3200° R (turbine-inlet temperature, about 3500° R).

The cooling losses for the engine with an effective gas temperature of 3200° R were analyzed and the approximate breakdown is as follows: The direct heat loss to the blades was 10.4 percent of the net brake horsepower; a substantial part of this energy would have been unavailable for work in any case because even in uncooled engines the thermal efficiency is only about 30 percent and much heat is thrown away, so that the term loss is misleading to some extent. The power required to pump the air from the turbine hub to the blade tips was 3.9 percent of the net brake horsepower; and the power in the main compressor required to increase the air to the required pressure was 10.9 percent of the net brake horsepower of the engine. The total pumping losses are thus about 15 percent, which would increase the specific fuel consumption from approximately 0.264 up to 0.293 pound per horsepower-hour although the specific output would remain very high, about four times that of current turbine-propeller engines.

Typical turbojet engine. - An analytical study was made of the possibility of cooling a typical uncooled turbojet engine. The method of analysis was similar to that previously given for the varying-design engines. The solid blades on the uncooled turbine were assumed to be replaced by air-cooled blades. Two configurations were studied: blades with seven fins, and blades with an insert similar to the blade shown in figure 5. Because the original uncooled turbine blade was so thin, a choking Mach number in the air passage was reached at an altitude of 20,000 feet. Consequently, altering the blade design, which increased the inside passage areas by 40 percent and reduced the number of blades from 120 to 100, was necessary. This alteration, of course, would necessitate a redesign of the blade profiles. The assumption was made that the blades were without shrouds. This assumption makes a large difference in required ratios of coolant flow to gas flow because shrouded blades are more highly stressed than unshrouded blades. With the altered blades, calculations of the ratio of coolant flow to gas flow required to maintain the same turbine-inlet temperature (2000° R) as currently obtained by the engine were made. The calculations were made for a range of altitude, blade life, and three materials. The materials were Cr-Mo-Va with about 5 percent of strategic alloys and 95 percent of iron, C-Mo with about 1 percent of strategic alloys, and SAE 1015 with no alloys.

Some results of the calculations are shown in figure 12. The curves in figure 12(a) show the increase in ratio of coolant flow to gas flow required by the rotor with finned blades as blade life based on stress-rupture properties is increased for the three materials. A 1000-hour life is usually assumed for airplane service but for missile applications a much shorter life could be assumed. The assumption of a certain life value does not mean that such a life will be obtained in practice. A high value is chosen because other stress factors that cannot be evaluated may appreciably reduce the assumed value. Small decreases in alloy content appreciably increase the ratio of coolant flow to gas flow required. In general, reduction from 5 to 0 percent increases the ratio of coolant flow to gas flow required from 1.5 to 2.5 at 100 hours of blade life. Reduction of life expectancy from 10,000 to 100 hours appreciably reduces the required ratio of coolant flow to gas flow. From these curves, the service requirement is evidently an important factor and therefore the alloy content should not be reduced too much. A very satisfying result shown in figure 12(a) is that for the altitude given, 40,000 feet, a dilution of only a little over 0.02 is required for 10,000 hours with Cr-Mo-Va.

The effect of altitude and blade configuration on ratio of coolant flow to gas flow is shown in figure 12(b) using Cr-Mo-Va blades with 1000-hour stress-rupture properties. For the blade with insert and the seven-fin blade, the ratio of coolant flow to gas flow increases to some extent as altitude increases. The ratios of coolant flow to gas flow with both blade configurations, however, are not excessive even at an altitude of 40,000 feet. The blade with the insert is less effective, requiring about twice the ratio required by the seven-fin blade. The nozzles or stator blades required a ratio of about 0.01.

Calculations of the percentage reduction in thrust of the uncooled engine due to losses associated with cooling were made. The calculations were made for a constant flight Mach number of about 0.8 and a range of altitudes. The losses include the power to pump the air through the turbine rotor and blades and the power of the compressor to compress the cooling air to the pressure required at the turbine-rotor inlet. The losses also include the effect of extracting heat from the working fluid while expanding through the turbine; this effect results in increased turbine pressure ratio. This increase in turn reduces the available pressure for expansion through the jet nozzle. The reduction of downstream gas temperature because of the mixing of the cooling air with the gas downstream of the turbine rotor is included. This decrease results in a reduction in jet velocity. The loss due to the air mixing with the gas is not included as it is an intangible loss. The reduction in fluid for producing work in the turbine is included. This reduction is caused by bleeding of air from the compressor for the cooling. Finally, the loss due to an increase in fuel-air ratio is included.

The results of these loss calculations are shown in figure 13 for the seven-fin blade and the blade with insert where the percentage loss in thrust is plotted against altitude. A general decrease in percentage loss in thrust occurs from sea level to 40,000 feet. The loss in thrust appreciably increases as ratio of coolant flow to gas flow is increased, from about 5 percent at a ratio of coolant flow to gas flow of 0.03 to about 16 percent at a ratio of 0.07 at sea level for the blade with insert. Although use of the seven-fin blade produces more loss in thrust than the blade with the insert at a given ratio, this difference is insignificant. Of significance is the requirement at an altitude of 40,000 feet (fig. 12(b)) of a ratio of only about 0.018 when the seven-fin blade is used, whereas about 0.034 is required when the blade with insert is used. Consequently, from figure 13 the loss in thrust would be only about 3.9 percent for the seven-fin blade as compared with about 6.5 percent for the blade with the insert.

These results were obtained for the condition of maintaining the turbine-inlet temperature at the same value as in the uncooled engine. Because of the losses, the cooled engine would consequently have a lower power output than the uncooled engine. In order to overcome these losses, which are not large, and to obtain the same power, the turbine-inlet temperature would have to be increased; an approximate value was determined as 1600° F for the seven-fin blade. This value would require the passing of more air than 1.8 percent at an altitude of 40,000 feet through the rotor blades.

This discussion of a typical engine was made to point out problems involved in applying cooling and the details of solution are not of particular importance.

The loss study showed that the two most important effects, especially for the blade with insert, are:

(a) Loss in turbine mass flow: For required compressor work, the turbine pressure ratio must be increased, resulting in lowered available jet-nozzle pressure ratio.

(b) Loss in total temperature of the gas in the tail pipe due to mixing of turbine coolant at 960° to 1160° R with the exhaust gases at 1660° to 1760° R: This loss lowers total enthalpy available at the jet nozzle.

The entire study of the typical turbojet engine was most encouraging. The ratios of coolant flow to gas flow required and the loss in thrust were not excessive, and the end conclusion that seems to be indicated, which is current jet-engine performance can be obtained when cooling is applied to blades made of nonstrategic materials, is of immense importance to national defense.

OBJECTIVES OF TURBINE-BLADE-COOLING RESEARCH

Because of the indicated improvement in specific thrust and, in some cases, specific fuel consumption due to increased turbine-inlet temperature and the probability that nonstrategic materials can be used, the NACA has been conducting a research program on turbine cooling to apply and check the theoretical results.

The general objectives of this research are:

(1) Provide knowledge such that turbines made of nonstrategic materials can be operated at gas temperatures as high or higher than those currently used

(2) Provide knowledge such that turbines can be operated at high gas temperatures

The detailed objectives of this research are:

(1) Provide analytical methods for accurately designing cooled turbines

(2) Provide analytical methods for accurately evaluating performance of cooled turbines and engines with such turbines

(3) Provide methods for conducting experiments and calculating data on cooled turbines in order to be able to evaluate the performance

(4) Provide experimental data, obtained on rigs and turbines, that are necessary to establish formulas for factors required in the methods of design and evaluation of cooled turbines for which no theory or inadequate theory are available. Some of these factors are: heat-transfer coefficients for the blades, both inside and outside; friction factors for coolant passages; fluid temperatures effecting heat transfer, and such factors as turbine efficiency, blade power, and turbine mechanical losses; that is, data to establish formulas for performance, heat transfer, and coolant-flow characteristics.

(5) Develop the theories by which the foregoing factors can be determined

(6) Explore and expedite methods of manufacture of special blade configurations

(7) Fabricate and operate actual cooled turbines

(8) Analytically explore the possibilities of cooling on engine and airplane performance considering all variables such as blade material, material life, blade configuration, service requirement, and so forth

(9) Expedite the accumulation of material properties such as fatigue, stress-rupture, endurance limits, and notch sensitivity at elevated temperatures

STATUS OF COOLED-TURBINE KNOWLEDGE AND INVESTIGATIONS

The status of cooled-turbine knowledge and investigations includes a discussion of the status of theoretical knowledge and experimental investigations, cooling-flow characteristics, turbine and engine performance, turbine performance characteristics, and engine performance characteristics.

Theoretical Knowledge of Heat Transfer

Allowable blade-temperature distributions. - In order to establish the allowable gas temperature at which a cooled gas turbine can operate for a given coolant flow or to establish the coolant flow required to operate at a given gas temperature, the allowable blade-temperature radial-distribution curve must be determined. A discussion of its use was given in the discussion of figure 3. In order to predict the operating life of a unit with a reasonable degree of accuracy and to keep the required coolant flow to a minimum, the allowable temperature of a turbine blade must reflect the actual physical properties and stress distribution of the blade as closely as possible. In analytical work to the present time, the centrifugal stress imposed on the blade has been used in conjunction with stress-rupture data to determine an allowable temperature-distribution curve, as previously explained. The characteristic of this curve is to show a lower allowable temperature at the blade root where centrifugal stress is highest. (See fig. 3.) Because thermal stresses are present in a cooled blade, a relation, as previously noted, is needed between thermal, centrifugal, and gas-bending stresses for a particular blade. An additional stress is the vibratory stress, which is caused by various factors discussed in CURRENT LIMITATIONS ON PERFORMANCE. The knowledge of blade stresses is extremely limited as are data on elevated-temperature endurance limits, and so forth, which are needed to calculate accurately the allowable-temperature-distribution curve.

A brief description of the current method of obtaining the allowable-temperature-distribution curve is as follows: The cross-sectional areas of the metal portion of the blade A_B from root to tip are plotted against the distance from the root. Small increments of rotor radius dr from the blade root to tip are taken and the stress at the tip calculated from

$$s = \frac{\rho_B \, (A_B)_{av} \, dr \, \omega^2 r}{144 \, A_B}$$

where A_B is the area at the base of increment and $(A_B)_{av}$ is the average cross-sectional area of the increment. For the next increment toward the root, the stress is calculated from

$$s = \frac{B + \rho_B \, (A_B)_{av} \, dr \, \omega^2 r}{144 \, A_B}$$

where all terms correspond to the increment under consideration except B. The term B equals the stress multiplied by the base area of the first increment. The stress of each remaining increment is determined in a similar manner, B in each case being the stress multiplied by the base area of the preceding increment. If the stress distribution of the blade is known, the allowable temperature distribution can be obtained from stress-rupture data for the material used in the blades. Usually the stress-rupture data are plotted as stress against temperature with life in hours as a parameter.

Actual blade-temperature distributions. - The theory for determining actual blade-temperature distributions is almost complete. Considerable work has been published by both the NACA and in Germany on this subject. The German work is summarized to some extent in reference 12. As previously stated in the discussion of the methods of analysis for comparing cooling methods, the temperature-distribution equations for air-cooling are derived in reference 6 and for water-cooling in references 9 to 11. The theories are complicated and involve assumptions; the details will not be discussed. It is sufficient to note that much theoretical information is available for calculating the actual temperature-distribution curve (solid line, fig. 3). The equations are long and the material must be put in more usable form for designers. The NACA is endeavoring to prepare charts from which blade temperatures can be determined for various blade configurations, cooling methods, and materials. The reduction of the results from complicated equations to chart form is uncertain.

The following example of such an equation for radial temperature distribution is for air-cooled blades based on one-dimensional theory:

$$\frac{T_{g,e} - T_{B,x}}{T_{g,e} - (T_{a,e})_h} = \left(\frac{1}{\lambda+1} + X - Y\right) e^{-Z} - \frac{\omega^2 w_a b}{gJH_o l_o \left[T_{g,e} - (T_{a,e})_h\right]} \frac{x}{b} - X + Y$$

where

$$Z = \frac{1}{\lambda+1} \frac{H_o l_o b}{w_a c_{p,a}} \frac{x}{b}$$

$$\lambda = \frac{H_o l_o}{H_1 \ l_1}$$

$$X = \frac{\omega^2 r_h w_a}{gJH_o l_o \left[T_{g,e} - (T_{a,e})_h\right]}$$

$$Y = \frac{\omega^2 w_a^2 (\lambda + 1) c_{p,a}}{gJH_o^2 l_o^2 \left[T_{g,e} - (T_{a,e})_h\right]}$$

The blade temperature depends on the effective fluid temperatures $T_{g,e}$ and $(T_{a,e})_h$ and the convection heat-transfer coefficients H_o and H_i. (Radiation was neglected in this derivation so that radiation coefficients do not appear. A discussion of this procedure is given in reference 6.) The blade temperature is consequently only as accurate as the accuracy of the effective fluid temperatures and the heat-transfer coefficients. These factors will be subsequently discussed.

General theory for convection heat-transfer coefficients. - All problems of convection heat transfer are problems of flow of heat through the boundary layer. Consequently, setting up the equations for the flow of fluid and the temperature variation in the boundary layer supposedly could establish the heat-transfer coefficient laws. The heat-transfer coefficient would be obtained from the slope of the fluid-temperature curve at the wall using the equation

$$H = \frac{-k_f (\partial T/\partial y)_{y=0}}{T_{f,0} - T_w} \tag{3}$$

Details of this relation are discussed in reference 13.

Dryden (reference 14) has set up such equations for both laminar and turbulent flows. For laminar flow, nine equations, five of which are differential equations, result with nine unknown terms. Up to the present time, little progress has been made in the solution of this formidable problem. Simple problems, such as for flow through a tube and across flat plates, with the assumption that the flow is one or two dimensional, and many other assumptions have given rise to a solution of the equations that fits experimental results fairly closely. For turbulent flow, the equations are more difficult; the fluctuating-velocity terms and their equations enter into the problem in addition to equations of the type for laminar flow. Consequently, the theory of heat transfer in eddying flow is unsatisfactory.

As in many other fields of technical physics where the differential equations governing the phenomena are such that solution is impractical, if not altogether impossible, experimental techniques aided by the method of dimensional analysis based on the π theorem have proved a valuable tool in the treatment of heat-transfer problems. By use of such an analysis, the Nusselt number, which involves the convection heat-transfer coefficient, can be shown to be a function of the parameters given in the following equation:

$$Nu = \psi\left[Re, Pr, Gr, M, \frac{k_f(T_x - T_f)J}{g\mu_f V_f^2}, \beta(T_x - T_f), \frac{D}{L}\right]$$

The case of natural convection for which $V_f = 0$ is first considered. The Reynolds number Re, Mach number M, which involves V_f, and the term $k_f(T_x - T_f)J/g\mu_f V_f^2$ are then absent and the equation becomes

$$Nu = \psi'\left[Pr, Gr, \beta\ (T_x - T_f), \frac{D}{L}\right]$$

The quantity $\beta(T_x - T_f)$ is a measure of the volume changes in the fluid but has considerable influence only in special problems, such as in the loss of heat from a hot wire of very small diameter. As a consequence, the equation for natural convection has generally been of the form

$$Nu = \psi''(Pr, Gr)$$

for a given D/L.

For forced convection, the natural-convection currents are usually negligible so that terms involving β and g have no influence on the heat transfer. The Grashof number Gr and $\beta(T_x - T_f)$ are then absent. It is also an experimental fact that H in the Nusselt number Nu is proportional to $(T_x - T_f)$. A term involving $(T_x - T_f)$ therefore cannot enter on the right side of the equation. Thus for forced convection, the equation reduces to

$$Nu = \psi''' (Re,Pr) \tag{4}$$

for a given D/L.

Experimental evidence indicates that the D/L term is generally negligible and the equations given for natural convection and forced convection have been applicable to most heat-transfer problems. The functions ψ'' and ψ''' must be experimentally determined.

Nothing has been said about the Mach number M in equation (4). In the past, flows have been at slow speed, which automatically eliminated M. As M increased, in the equation for H (equation (3)) the temperature $T_{f,0}$ had to be replaced with the effective fluid temperature $T_{f,e}$ in order to eliminate M from the equations. The relation between the effective temperature and the static and relative total temperatures was previously given (equation (1)). The effective temperature of the fluid is the temperature the body will assume, for the same fluid conditions, when it is unheated and uncooled.

Turbine-blade "gas-to-blade" convection coefficients. - An equation of the form of equation (4) has successfully been used to correlate heat-transfer data when fluids are flowing around pipes, across tube banks, and around streamline struts for design angle of attack. The Nusselt number is based on average heat-transfer coefficients. The application of one function ψ''' to all turbine-blade shapes, ranging from impulse blades to reaction blades, and for all angles of attack is optimistic. Rather, it is to be expected that for each blade shape and for each angle of attack of the flow a different function will prevail. Also the function will probably change with or without nozzles upstream of the rotor blades. A plot of Nu divided by Pr raised to some power against Re will therefore show separate curves for each blade shape and angle of attack.

The reason for the inability to correlate all the average heat-transfer coefficients for blades on one curve, as is done for tubes, becomes evident when figure 14 is considered. This schematic sketch illustrates the flow that may exist around a turbine blade at design angle of attack. Studies that have been made indicate that the flow in the boundary layer on the concave side of the blade will be laminar. On the convex side, however, a transition from laminar to turbulent flow occurs at some point along the surface. The heat transfer for laminar flow is much less than for turbulent flow so that the average heat transfer for the blade will change appreciably depending on the extent of the laminar and turbulent regions. Blade shape and angle of attack can appreciably change this transition point. Another factor that has become evident concerning heat transfer and flow is that the decreasing ratio of the temperature of the surface to the fluid temperature outside the boundary layer increases the stability of the boundary layer, delaying the separation point of the fluid from the surface and the transition from laminar to turbulent flow. Whether the fluid separates or not will, of course, greatly affect the heat transfer. The temperature ratio mentioned also seems to be another factor that enters into equation (4) and affects the heat transfer even though the fluid does not separate. A high heat-transfer rate from the blade to the coolant, which removes a large amount of energy from the boundary layer thus lowering its pressure, can cause separation to occur sooner than a low heat-transfer rate. All these factors necessitate development of theoretical means for determining local values of heat transfer around the blades for any blade shape and angle of attack that can be averaged to obtain points on a curve, which follows a law such as expressed by equation (4). Only by such means can a reduction be made in the tremendous amount of experimental research that would otherwise be required to establish curves for all the variations that exist in blade shape, and so forth.

The ultimate approach to these heat-transfer laws is to derive methods whereby the heat-transfer parameters can be independently determined from aerodynamic considerations such as blade-profile shape, relative flow angles and Mach number, velocity distribution, and the laws governing the development of the boundary layer because the boundary layer is controlling. The two general methods of solving boundary-layer problems are:

(1) Solving the boundary-layer differential equations of continuity, motion, and energy

(2) Solving the boundary-layer momentum equation

In reference 15 are listed solutions by various investigators using the first method for incompressible flow and nonvariation of fluid properties in the laminar boundary layer, that is, density, viscosity, and specific heat are constant from the wall to the stream. Some of these investigators assumed a pressure gradient of zero parallel to the surface (flat plate) but partly accounted for the pressure gradient by inserting local velocity in the equation for heat-transfer coefficient. Others, after the assumption of a pressure gradient of zero, introduced a boundary-layer thickness that embodied the velocity variation along the surface. Reference 15 presents a solution based on a combination of both methods wherein the pressure gradient was neglected in the energy equation but considered in the momentum equation. Fluid-property variation and compressibility effects were neglected. For high heat transfer, the property variation is significant but for a Mach number less than 0.2 neglect of compressibility is not serious.

The first method is used in reference 16 and includes compressibility effects and fluid-property variation but neglects the effects of a pressure gradient. This method is also used in reference 17 and includes the pressure terms but excludes fluid-property variation and compressibility effects.

The work of Crocco and Conforto, which is similar to reference 16, is utilized in reference 13 and a method for correlating heat transfer is obtained by basing the temperature for fluid-property evaluation on a function of Mach number and temperature ratio. This method, however, is for a pressure gradient of zero.

The effects of fluid-property variation and the pressure gradient have not been included in the same analysis, insofar as can be determined. An investigation utilizing reference 16 is being made by the NACA to account for these two significant features of the heat transfer with a laminar boundary layer.

With the use of method (1), the Reynolds analogy (Prandtl number, 1 and pressure gradient, 0) was employed in the turbulent case that resulted in the dependence of the heat-transfer coefficient on the friction factor. Modifications of this analogy to apply for other values of Prandtl number and pressure gradient have been made. The momentum equation (method (2)) was used by some investigators. The results of the use of both of these methods are shown in reference 15. Some of the equations for heat-transfer coefficient account for a nonunit Prandtl number, whereas all the equations account for the pressure gradient in some manner. One of the basic assumptions is neglect of the dissipation term.

In reference 18 the first method and the Reynolds analogy are used to obtain an equation for heat transfer including dissipation. The variance between this analysis and that where dissipation was neglected does not seem to be large.

Some equations for the turbulent boundary layer derived from pipe-flow theory, which may be applicable if a turbulent boundary layer exists on a cooled turbine blade, are presented in reference 13.

From the stability calculations (reference 19), apparently a laminar boundary layer might well exist for a cooled turbine blade. Where a favorable pressure gradient is obtained, this existence would be true and even with a slightly adverse pressure gradient the laminar region may still prevail when high cooling rates are present.

Much work remains in order to arrive at a theory that will be applicable to the conditions existing when heat is transferred from hot gases to cooled turbine blades. Whether a satisfactory theory, which will agree closely with experimental results, can be evolved is currently unanswerable.

Radiation transfer of heat to blades. - Radiation from turbine walls to stator and rotor blades and from stator blades to rotor blades will probably have a prominent part in the heat-transfer problem of the cooled turbine. The rate of radiant heat transfer depends on the well-known formula that the heat rate is proportional to the difference between the hot body temperature raised to the fourth power and the cold body temperature raised to the same power, and to other factors, including a term called the geometry factor. The equation for radiant heat transfer from a small finite area of turbine shroud to a small finite area of the blade is

$$Q_r = \sigma \alpha_B \alpha_W \left(T_W^{\ 4} - T_B^{\ 4}\right) \iint \frac{\cos \theta_B \cos \theta_W \, dA_B \, dA_W}{\pi r^2} \tag{5}$$

The quantity under the integral signs is equal to the geometry factor multiplied by the small finite area on the blade surface being evaluated. Although geometry factors exist for tubes and other simple bodies, none exist for the complicated patterns such as those formed between walls and blades and stator and rotor blades. In addition, no theory exists for their determination. The very tedious complex operation must therefore be performed of finding the individual factors for each small area on the blade and for each small area that this blade "sees" by using equation (5). Large wood models, about 20 times the size of the blade cascade must be made for accuracy. A device called a mechanical integrator described in reference 20 is used to integrate graphically the portion of the equation under the integral signs in equation (5). The method of obtaining the average geometry factor for the blade from such measurements is given in reference 21. The radiant-heat aspect has received little attention and much work is required to develop theories for calculating radiation geometry factors for complicated shapes.

Turbine-blade convective heat transfer to coolant. - The flow of the coolant through the inside of the blade can be thought of as flow of a fluid through a tube where the tube has odd shapes as in the blade. The tube could be formed by the blade walls in a hollow blade, by the blade walls and insert outer surface, if an insert is used, which would result in a tube with a shape somewhat like an annulus, or by the blade walls and fin surfaces if fins are placed in the passage. These shapes are illustrated in the sketches in figure 5. A formula such as represented by equation (4), which is applicable to flow through tubes, might therefore be supposed applicable for convection heat-transfer coefficients on the inside of blades, the tube shape of which may vary greatly. For the term D in the Nusselt and Reynolds numbers, the average hydraulic diameter of the passage D_h or passages (if fins are used) would then be used as is the practice for heat transfer in rectangular tubes.

For fully developed turbulent flow in pipes, it has been ascertained (reference 3, p. 168) that the formula for heat transfer is

$$Nu = 0.023\ (Re)^{0.8}(Pr)^{0.4} \tag{6}$$

in which the fluid properties were based on the bulk temperature of the fluid. Evidence now exists from tests in tubes (reference 22) where the ratio of wall to stream temperature is far from one that, as previously noted, an equation similar to equation (6) will not apply over the entire temperature-ratio range. An equation very similar to equation (6) was determined to apply if the properties of the fluid including the density are based on wall temperatures.

An equation similar in form to equation (6) has been found to apply to convection heat transfer in annular spaces with the diameter replaced by the hydraulic diameter of the annular space (references 23 to 25 and reference 3, p. 200). In addition, according to reference 24, the constant 0.023 has to be multiplied by a factor dependent on the ratio of the diameters of the inner and outer pipes forming the annulus. In all cases, the ratio of metal-surface temperature to fluid-stream temperature was not extreme.

Another factor that affects the heat transfer in tubes is the distance from the tube inlet. Generally, at the tube inlet the fluid velocity is uniform across the tube and no boundary layer exists, as illustrated in figure 15(a). As the fluid proceeds, boundary layers, usually laminar, build up along the walls until, if the Reynolds number is low, fully developed laminar flow with a parabolic velocity distribution exists in the tube, as indicated in the velocity-distribution diagram in figure 15(a). If the Reynolds number in the main stream is high, the laminar layer will change to a turbulent layer at some transition point and finally grow to a point where fully developed turbulent flow will exist in the tube some distance from the inlet. The heat-transfer coefficient varies, as indicated in figure 15(a), to the point where fully developed laminar or turbulent flow exists. Its variation will also depend on the Reynolds number of the fluid. Consequently, formulas such as equation (6) apply to only a limited condition, for fully developed flow-velocity distribution and not to the transition section. Some aspects of effect of inlet length are discussed in more detail in reference 26.

In the cooled-turbine-blade inside passages, inlet effects on heat transfer on the inside may occur depending on the manner in which the air is introduced at the blade root. If the air enters the blade from a chamber that is rather large, the flow may be somewhat similar to that described downstream of the tube in the previous paragraph. Other arrangements may cause a flow that has appreciable boundary-layer thickness at the blade root. The annular-space discussion points out that in trying to correlate heat transfer in hollow blades with that in hollow blades with an insert, which forms a sort of annulus, a factor based on the annulus dimensions may have to be used as a multiplying factor in the equation for the hollow blade before agreement is obtained with blade-with-insert results.

No known data exist on heat-transfer coefficients inside turbine blades. It is impractical to expect to obtain data inside blades of ordinary commercial turbines that will (1) give formulas for variation of heat transfer along the blade span, (2) give factors that will correlate different inside geometries of the tube passages, and (3) evaluate the type of flow existing. Only an average coefficient for a particular configuration and detailed data on local coefficients using large turbine rigs and static cascade rigs can be expected. When such detailed data are obtained, it is hoped that methods can be evolved to obtain integrated inside coefficients that will check the average values, which will be obtained on typical turbines.

As pointed out, the foregoing method is recommended on the premise that Reynolds and Prandtl numbers are the primary factors affecting heat transfer. This premise is usually valid for forced convection. As previously noted, the heat-transfer equation derived by dimensional analysis has many parameters that affect heat transfer. The absence of most of these parameters, such as that for forced convection, is due to the negligible effect they have for the existing type of flow. In the cooled turbine, especially in the coolant passage, the Grashof number may also be a factor affecting heat transfer because the Grashof number for ordinary natural-convection processes contains the gravity term g. In a turbine, however, when the coolant passage is considered, g must be replaced by a term, which may be from 10,000 to 50,000 times the ordinary value of g depending on the turbine speed. As a consequence, the Grashof number reaches values that are appreciable and in some cases, such as when the water holes are not drilled through, may overwhelm the Reynolds number effect on heat transfer and therefore equation (6) would not represent the experimental findings.

A turbine has been operated in Germany on a natural-convection principle using water as the coolant. The holes are not drilled through, as shown in the sketch in figure 15(b) and the water flows up through the center of the holes and down the outside of the holes. The NACA also has two water-cooled turbine wheels using this cooling principle.

Effective fluid temperatures. - The heat-transfer coefficient is affected by many parameters, as brought out in the discussion of the general theory of convective heat transfer. These parameters can be determined by means of the dimensional theory. One such parameter is the Mach number of the fluid. As mentioned, depending on conditions of flow, some parameters have negligible effect and the experiments result in formulas that do not contain these parameters. In the past, a great amount of research on heat transfer was conducted at low Mach numbers and this parameter did not affect results. Consequently, although from dimensional theory Mach number would appear in the heat-transfer-coefficient equation, in experimental formulas it did not appear. Also, the choice of temperature to use for fluid temperature in evaluating the heat-transfer coefficient H did not appear to affect H. Consequently, stream static temperature was usually used. As fluid speeds increased, however, the coefficient H was affected by Mach number if H was determined using stream static temperature.

Several theoretical analyses and some experiments have recently been made from which it has been determined that the normal type of formula, in which Nusselt number is a function of Reynolds number and Prandtl number, is applicable to high fluid speeds if the heat-transfer coefficient is based on the effective fluid temperature (references 4, 13, 18, and 27 to 32). That is, the Mach number parameter drops out of the equation for Nusselt number on this basis. The effective fluid temperature is the temperature affecting heat transfer and is equal to the temperature the body surface would assume if it were thermally insulated, no heating or cooling, under the same fluid conditions for which the heat-transfer coefficient is being determined. The effective fluid temperature is thus the adiabatic surface temperature. A discussion with diagrams of the formulas for H on different bases and on adiabatic surface temperature is given in reference 13.

The adiabatic temperature of a body such as a turbine blade, like that for a thermocouple, can be determined using formulas similar to equation (1) from the total and static temperatures of the fluid flowing around or through the body and from a recovery factor if they are known.

A general summary of the work done by many investigators to establish the formulas for the recovery factor Λ for several bodies is given in reference 13. In general, the conclusion has been made that Prandtl and Mach numbers are the paramount parameters affecting Λ. Some theory and experiments on flat plates indicate that Λ varies as $\sqrt{Pr}$. Investigations on a cascade of turbine blades (reference 27) shows that

$$\frac{\Lambda}{\sqrt{Pr}} = f(M)$$

No analytical solutions for determining the effective fluid temperatures outside and in the cooling passages of turbine blades are available. The solution of the boundary-layer equations, which will lead to methods for calculating the characteristics of the fluid in the boundary layer is the first step in the analytical determination of effective fluid temperatures. Recovery factors, and thus effective fluid temperatures, have been analytically evaluated for flat plates and for wedges, the wedge evaluation being for laminar flow. This work is described in reference 13. These methods of evaluation can serve as guides for the turbine-blade recovery-factor analytical evaluations.

Experimental Investigations on Turbine-Blade Heat Transfer

Requirements for apparatus. - Because of the lack of analytical methods to calculate various heat-transfer factors required in the evaluation and design of cooled turbines, experimental procedures and apparatus and experimental data obtained therefrom are required to obtain the factors. In experiments, both the static cascade and the rotating cascade, which is used in both turbine rigs and turbines, are required.

In experimental work, the stationary cascade is an essential part of a research program leading to the development of reliable methods and physical relations for rational design of cooled-turbine engines. An adequate understanding of the advantages and the limitations of blade cooling requires research that cannot very well be conducted on a full-scale turbine. Blade-cooling investigations in an actual turbine suffer four severe limitations:

(1) Inaccessibility for necessary instrumentation

(2) Impossibility of isolating and individually controlling convection, radiation, and conduction without exceeding construction and operating restrictions

(3) Restriction of maximum gas temperature in existing machines, which effectively prevents rapid exploration of cooling processes at the high temperature levels

(4) Impossibility of rapidly investigating various cooling methods, blade configurations, and so forth

Both basic and empirical data can be obtained with static cascade rigs. The distinction between basic and empirical data depends on the method of approach in layout and operation of the rig. Basic data for the heat-transfer equations may be obtained by breaking down the problem to eliminate or to control each effect and thus determine true convection and radiation coefficients. The analytical methods are then used to combine the basic elements for design purposes. The empirical data would be obtained where the rig was designed and operated to reproduce the conditions of an actual turbine stage. The data would consist of temperature-response relations resulting from changes in inlet temperature, radiation-source temperature, angle of attack, and so forth.

With static cascade rigs, the effects of all or some of the following factors on heat-transfer coefficients and blade temperatures can be studied:

(1) Gas-temperature variation

(2) Coolant-temperature variation

(3) Coolant flow

(4) Gas flow

(5) Rim, air, liquid, film, and boundary-layer cooling

(6) Blade configuration

(7) Blade size (blades range in size from 3/4- to 6-in. span, for example)

(8) Blade angle of attack

(9) Radiation heat transfer

(10) Convection heat transfer

(11) Ceramic coatings

(12) Coolant Mach, Reynolds, and Prandtl numbers

(13) Gas Mach, Reynolds, and Prandtl numbers

(14) Blade material

The rigs should be so constructed that either local heat-transfer coefficients around the blades, average coefficients, or both can be obtained. In general, much of the basic data should be obtained using hot and cold air or cold water (these data should be checked at high temperatures with combustion gases) in order to isolate convection effects, to overcome the lack of knowledge concerning combustion-gas properties, and to eliminate as much as possible the effects of carbon deposits on the blades and the possibility of incompletely burned gases. On the basis of cascade aerodynamic data, the number of blades should be as large as is permitted by the facilities. Heat dams should be provided to eliminate heat losses from the blade ends as much as possible. On the coolant side, the blades should be instrumented in such a manner that inlet effects on the heat-transfer coefficient can be obtained. The coolant flow, of course, is not fully developed into turbulent or laminar flow until it proceeds some distance through the blade coolant passages. Thermocouples should be so located on the blades that the spanwise distribution around the periphery is obtained, as well as a detailed peripheral distribution along the span. In addition, the coolant passage should be fully instrumented along the camber line of the blade at about two stations representing inlet and outlet of the instrumented part of the blade. An additional probe should be used in conjunction with these surveys to determine the conditions in the passage between the blades.

Boundary-layer temperatures and velocities are required for a full solution of the cooling problem. The practical size of static rigs precludes use of pressure tubes in the boundary layer; multiplication of the estimated boundary-layer thickness of the gas film around the blades by a factor of at least 4 would permit such use. Near leading edges, hot-wire probes are also severely limited in application. Because of aerodynamic loading at velocities in excess of 100 feet per second, wire length-diameter ratios must be restricted to values of the order of 60. End-conduction effects are large and difficult to calculate accurately because of uncertainties in flow and temperature distributions. Other effects too numerous to mention affect the accuracy of measurement. Some simple calculations show that interferometric methods can be used to obtain boundary-layer measurements in static rigs of the size that is practical. Excellent photographs of flow around turbine-blade profiles were obtained (reference 33) with an interferometer and indicate that such methods can be used to determine boundary-layer characteristics.

The advantage of being able to apply data from static cascade rigs, which are simple to construct and to operate, to turbine design is evident. The applicability of the data is determined by checking against data from rotating blade cascades. Rotating blade cascades are expected to yield considerable data on such factors as boundary-layer flow with centrifugal forces and under radial pressure variations on both inside and outside surfaces of the blades, natural convection effects tending to augment the internal heat-transfer coefficient, compressibility temperature-rise effects in cooling-air passages, actual losses incident to pumping air through blades, and relative losses of various discharge methods (that is, from tip, with and without controlled vector, or into boundary layer). None of these vital factors can be handled in static cascades.

A turbine rig, including the blades, should be thoroughly instrumented with both thermocouples and pressure tubes. Measurements should be taken in the blade-cooling passages to determine flow and heat-transfer coefficients. The thermocouple and pressure leads from rotating parts will probably have to run radially inward between the disks into a hollow instrument shaft, which will also serve to drive pressure and temperature pickups.

Description of NACA apparatus. - Currently the NACA has at the Lewis laboratory four static cascade rigs, two special static rigs, two turbine rigs, and four cooled turbines in the process of design, fabrication, or operation or a total of 12 pieces of apparatus to study the turbine-cooling problem.

A typical static cascade is shown in figure 16. This cascade was one of the earlier models and four blades were used. Because of the great quantities of air needed and the large apparatus required to heat this air to obtain high fluid Mach numbers between the blades, the number was restricted to four although more blades should be used if possible. The blades are hollow so that cooling air can be passed through them. Because of their large size, about 6-inch span and 5-inch chord, they are fully instrumented to obtain complete data on hot-gas, cooling-air, and blade-temperature and -pressure characteristics. The mass of instrumentation is evident in the figure. Facilities are available for both hot-combustion-gas and hot-air determinations. This rig is used to obtain local heat-transfer coefficients around the outside of the blade and in the blade-cooling passage.

One of the special static rigs is shown in figure 17. This rig is used to study the boundary layer around a cooled turbine blade; the data so obtained are used to verify equations giving the characteristics of the boundary if possible. When the boundary-layer characteristics are known, the local outside heat-transfer coefficient, the recovery factor, and the effective gas temperature can be determined as previously explained. The rig, as shown, has a movable water-cooled symmetrical blade, which can be rotated or moved across the tunnel section. The vertical walls of the rig are also movable, rotate about a pivot, and have a special shape. By movement of the walls and the blade, various pressure distributions around the blade corresponding to typical pressure distributions of various shaped turbine blades can be set up. The blade is instrumented with thermocouples and pressure taps and a hot-wire anemometer probe is used to obtain the temperatures and the velocities of the fluid in the boundary layer. This rig is constructed but more development of the hot-wire anemometer is required in order that it can satisfactorily operate in the high-temperature, high-velocity field.

The interferometer offers another method to determine the characteristics of the boundary layer around cooled turbine blades. With this instrument, a picture of the flow or "fringe patterns" are obtained. The deviation of each line in the fringe pattern from a point on the line outside the boundary layer is proportional to the change in density. If the static pressure in the boundary layer normal to the surface is assumed constant, as is always done currently, this deviation is then proportional to the temperature change. A graphical measurement of the deviation of each line on the interferometer pattern from a point outside the boundary layer and the measurement of temperature with a thermocouple at this point will provide data on the boundary-layer temperature variation. The other special static rig previously mentioned (fig. 18) is one with which such data can be obtained. It is a wooden replica of the rig in figure 17 except the walls and the blade cannot move. The positions of the blade and the walls are such that the pressure distribution is typical of that of turbine blades currently in use. The blade is made of Lucite and is cooled by ethylene glycol, which can be reduced in temperature to -40° F. The coolant manifolds on each end of the blade are indicated in figure 18. They obstruct the view of a small part of the leading edge of the blade but were made as small as possible. Ordinary air at room temperature is passed around the blade. Optical-glass windows are placed at the ends of the blade. The interferometer photographs are taken through these windows.

Verification of equations for blade-temperature distribution. - The investigations of turbines and turbine rotors have not progressed to a point where checks of the equations developed for calculating blade-temperature distributions can be made. Experiments on a static cascade of solid blades that were heated at the root, however, have been conducted (reference 27). Cold air was passed around the blades and the heat-transfer coefficients from the air to the blades were determined. Experiments were then conducted at conditions very different from those at which the heat-transfer coefficients were obtained and the blade temperatures were measured from root to tip. The blade-temperature distribution was then calculated using an equation previously developed. The comparison of the experimental and theoretical temperature distribution is shown in figure 19. The difference between the blade temperature and the effective fluid temperature is plotted against the distance along the blade measured from the root. The maximum error is only 10° F at a blade temperature of 1507° R or about 1 percent. This comparison of results using a blade-temperature-distribution equation with experimental results indicates that the theory is probably adequate.

Experimental results on blade outside heat-transfer coefficients. - Heat transfer from solid blades in a static cascade heated at the root to air flowing around the blades is discussed in reference 27. Data from which heat-transfer coefficients from hot air to the same cascade of solid blades cooled at the roots have been calculated have also been obtained at the Lewis laboratory but are unpublished. The blades were of the impulse type. The gas temperature in the cooled-blade investigations was about 760° R. Additional air heaters have been added to the setup in order that experiments at gas temperatures of 1060° R can be made. Combustion-gas experiments will also be made.

Tests were made in Germany (reference 34) on a static cascade of impulse blades, which were electrically heated, to determine heat-transfer coefficients from the blades to cold air flowing around the blades. The blades had a solidity of 1.33 and a turning angle of 102° as compared with a solidity of 1.92 and a turning angle of 104° for the blades in reference 27. Experiments have also been conducted on a static cascade of water-cooled blades (reference 35). The blades had an insert somewhat similar to that in the sketch shown in figure 5. The fluid passing around the blades was hot air for low-temperature runs and combustion gases for high-temperature runs. The greatest temperature of the gas recorded was approximately 1200° R. These experiments were conducted on one set of blades, which were first oriented in the rig such that a cascade of impulse blades resulted. The orientation was then changed so that a cascade of reaction blades resulted. No blade-temperature measurements were made with thermocouples for simplification purposes. The blade temperatures were calculated by a method based on some assumptions. The recovery factor, from which the fluid temperature effecting heat transfer was obtained, was also assumed.

A comparison of the NACA data on the heated and the cooled cascade of solid blades is shown in figure 20(a). The Nusselt number, which involves the heat-transfer coefficient between the blades and the fluid around the blades, divided by Prandtl number to the one-third power is plotted against Reynolds number of the flow. In figure 20(a), the properties of the fluid were based on the film temperature, which is a mean of the average blade temperature and the bulk temperature of the fluid. The mass flow $\rho_g W_g$ is the mass rate of flow of the fluid divided by the cross-sectional area of the inlet section immediately upstream of the cascade. The characteristic dimension in the Reynolds and Nusselt numbers is taken as the perimeter of the blade divided by π. A difference occurs in the results of the heated and cooled blades (fig. 20(a)).

Because of this difference, the fluid properties including density were next based on the blade temperatures. The velocity of the fluid immediately upstream of the cascade was still used. The comparison of the heated- and cooled-blade results on this basis is shown in figure 20(b). The two sets of results are brought closer together, the average difference being decreased from about 11 to 5 percent by changing from the film-temperature basis to the blade-temperature basis. This result seems to confirm the conclusions of other investigations (reference 22) that better correlation of results are obtained when the surface temperatures are used to obtain fluid properties.

A comparison of British and NACA data is shown in figure 21(a). The properties of the fluid, except density, are based on blade temperatures. The density is based on a mean of the effective gas and blade-wall temperatures and on an average of the inlet and outlet pressures. The velocity is that at the cascade outlet. Comparison of results on this basis was necessary because lack of information in the British report prevented changing to any other basis. The impulse-blade results show a difference of about 20 percent. The reaction-blade results are definitely lower than the impulse-blade results.

A comparison of NACA and German results are shown in figure 21(b). All results are for impulse blades. The greatest difference in the Nusselt number is about 30 percent. In this plot, the gas properties were based on blade-wall temperatures and the density and the velocity were the values immediately upstream of the cascade. Although reference 35 shows a comparison of British data with German data, the NACA has been unable to find data in the German reports that can put correlation on an outlet-velocity basis.

The results of figure 21 confirm the discussion of a need for a boundary-layer theory for determining local coefficients that can be averaged to obtain curves as shown.

Radiation coefficients. - No data on radiation coefficients for blades have been obtained. All investigations have been made at temperatures low enough so that convection was the predominating cause of heat transfer. This procedure is purposely used so that the laws of convection heat transfer can be established without the complication of radiation being encountered.

When investigations are conducted at high temperatures where radiation heat transfer cannot be neglected without introducing large errors, the geometry factor must be known for a particular blade installation. An integrator for determining the geometry factor is shown in figure 22. The center of the round base is placed on the blade at the point being considered and the carriage and the light-beam barrel are moved around; the light beam traces the area "seen" by the point considered while the pencil on the bottom of the tube holding the light source scribes a diagram on the chart at the base. The integrator must be used on an enlarged model of the blades being considered because of the size of the integrator. The diameter of the base of the integrator is 7 inches. The method of surveying the area is illustrated in figure 23.

Inside convection heat-transfer coefficients. - No known data are available on the convection heat-transfer coefficients from the blade to the coolant. Of course, ample provision for obtaining such data is built into many of the rigs and should be available shortly.

Recovery factors for fluid outside blades. - Some data have been obtained at the NACA Lewis laboratory (reference 27) and in Germany (reference 36) on recovery factors outside blades in static cascades. From these data, the effective gas temperature can be determined with equation (1). The results are shown in figure 24. The recovery factor is plotted against outlet Mach number. In all cases, the recovery factor increases somewhat with increase in outlet Mach number. The NACA data show an increase from about 0.8 to about 0.9 at the choking Mach number.

Recovery factor for coolant in blade passages. - No data exist on recovery factors from which the effective coolant temperatures inside turbine-blade passages can be obtained. Data are scarce even for flow through tubes. A curve from reference 30 on recovery factor of fluid flowing in tubes is shown in figure 24. This curve also shows an increase with increase in Mach number. In this case, the local recovery factor at various positions along the pipe were plotted against the local Mach number for various air-flow rates. The flow rate itself had no effect on the recovery factor; it affected the factor only through its effect on Mach number. Until more data become available, these are the best data that can be given for the inside recovery factors.

The discussion has been primarily concerned with the heat-transfer relations that occur in the over-all turbine-blade cooling problem. Of all the factors involved, the knowledge of heat transfer for the blade profile and the inside coolant passage is most important. Many other aspects are to be considered, which involve flow over the blades and through the coolant passage and the performance variables associated with the turbine as a component of an engine. The performance variables include consideration of the thermodynamic effects and mechanical losses inherent in removing heat from the cycle and circulating the coolant through the blading. This report is not intended to present adequately the status of the knowledge on each of these secondary although very important aspects, but the remaining sections of the report will outline the problem and indicate the method of attack, which is being followed.

Cooling-Flow Characteristics

In order to evaluate the effects of blade cooling on the performance of gas turbines, the flow characteristics of the coolant within the blade coolant passages must be determined. Knowledge of pressure, temperature, and velocity of the coolant along the passage is necessary to determine heat flow, blade temperature, inside heat-transfer coefficient, and cooling losses. The cooling losses include the work of pumping the coolant through the entire cooling system and the nonmechanical losses due to heat transfer and mixing, which involve entropy increases.

Air-cooling theory. - The analysis of flow in air-cooled blades must be based on the flow of a compressible fluid through a passage rotating at a high angular velocity under the combined effects of area change, wall friction, and a high rate of heat transfer. Until recently, analyses have been available for only such a flow in stationary passages. The NACA is therefore undertaking the theoretical study of the one-dimensional flow of gases in rotating passages.

Although some experimental work has been done by the NACA and other research organizations on the correlation of the friction coefficient as affected by high rates of heat transfer, insufficient data are available to permit the recommendation of a reliable correlation for nonisothermal friction coefficients in blade-coolant passages. Because of this lack of experimental data, the wall friction coefficients in blade-coolant passages must be estimated from the data for isothermal friction coefficients in smooth tubes.

Water-cooling theory. - Water may be circulated in the turbine-blade coolant passages by either forced or natural convection. The analysis of the flow of water by forced convection in the blade-coolant passages presents no particular difficulty and is, in fact, very much simpler than the analysis required for air cooling. Just as in air cooling, however, isothermal coefficients of friction for tubes must be employed in the analysis although high rates of heat transfer exist.

The mechanism of the flow of a liquid in the coolant passage by natural convection is not well understood although it is believed to be similar to the Grashof number effect in vertical heated plates. In the simplest theory of this type of flow, the cooler liquid is postulated to flow through the core of the blade-coolant passage to the blind end of the passage at the blade tip and then to return along the wall of the passage where the coolant is heated under the influence of a high-centrifugal acceleration. Actually, the flow is probably complicated by the existence of small secondary convective circuits along the entire length of the passage. This complication may result in breakdown of the circulation near the tip of the blade where very small passages of large length-diameter ratio are required by the aerodynamic design of the blade. The University of Delaware, under contract with the Office of Naval Research, proposes the study of natural-convection flow in stationary vertical tubes for both gases and liquids. The research will be guided to some extent by the work begun during World War II by German scientists on the study of natural-convection cooling of turbine blades.

The NACA has investigated a steel rotor that embodies natural-convection circulation of the liquid coolant. The data reveal nothing concerning the nature of the coolant-flow phenomena because the passages in turbine blades are too small to insert the proper measuring instruments. It is hoped that some knowledge of the non-isothermal friction coefficient can be obtained from power-loss investigations on this turbine wheel.

Turbine and Engine Performance

As previously indicated, one method of approach to turbine cooling is actual construction and operation of experimental machines on the basis of available data to evaluate performance, losses, thermodynamic effects, and operating problems. The final aspect is to combine all this knowledge and to evaluate the turbine as an engine component in terms of engine performance.

Operating investigations of aluminum water-cooled turbine (forced-convection principle). - A turbine rotor was built of 14S-T aluminum alloy with a tip diameter of 12.06 inches, a root diameter of 9.75 inches, and 50 impulse blades to be operated at a maximum tip speed of 1000 feet per second. As shown in figure 25, the blades and the disk are integral and, in order to facilitate machining, the blades were untapered and cut as a series of planes and cylinders. Into each blade were drilled four radial coolant holes and two transfer holes to make possible a forced flow through the blades. The outside ends of the 300 coolant holes were sealed with screwed-in plugs. The cooling water entered the turbine rotor at the center and flowed radially outward in the space between the rotor and the baffle plate, a portion of which is shown in the cross section in figure 25(b), through the two coolant holes nearest the leading edge of the blade, across the tip of the blade through the transfer holes, radially inward through the two holes nearest the blade trailing edge, and out the rotor through the axial-discharge holes.

The instrumentation provided for the preliminary runs was not extensive because these runs were primarily intended to indicate the practicability of operation with a liquid-cooled turbine of high-conductivity material. Briefly summarized, the operating results are as follows: The turbine was operated for a total of 92 hours of which more than 30 hours were at turbine-inlet temperatures between 2060° R and 2560° R. All operation was conducted at atmospheric inlet pressure and at several pressure ratios up to 1.8. In order to provide an additional factor of safety during turbine operation, the water outlet temperature was never permitted to exceed 610° R. It is therefore probable that an excessive coolant flow was used at all the points at which this turbine has been operated. These conditions, although not strictly representative of normal turbine operation, were maintained in order to safeguard the turbine during the preliminary runs. After the 92 hours of operation, a method of obtaining rotor-blade temperatures was made available and thermocouples installed in the blades indicated that at a rotor speed of 5000 rpm, a tip speed of 263 feet per second, and an inlet-gas temperature of 2560° R, the maximum blade-temperature was 744° R. This temperature occurred at the blade leading edge at a point midway along the span, and all the blade temperatures indicated were between this temperature and 640° R. During the runs in which the blade temperatures were obtained, the turbine was again operated at atmospheric inlet pressure but at extremely low pressure ratios. The turbine rig in operation at these conditions is shown in figure 26. The white hotness of the inlet duct and turbine shroud is visible in the photograph. It is of interest to note that a turbine wheel of aluminum can be operated at such high inlet-gas temperatures.

The mechanical method of sealing the ends of the coolant holes with screwed-in plugs appears to be practical inasmuch as no leaks developed around any of the 300 plugs although calculated pressures of 3400 pounds per square inch were indicated at the plugged surfaces. There was no evidence of oxidation, erosion, or blade elongation and no indication that any more than a small part of the turbine useful life was consumed. A complete report of these first operating investigations run on this turbine is found in reference 37.

Operating tests of steel water-cooled turbine (natural-convection principle). - A turbine rotor has been built of AISI 403 steel with a tip diameter of 13.88 inches, a root diameter of 9.0 inches, and 31 reaction blades. The turbine is designed to operate at a maximum mass flow of 14 pounds a second at a pressure ratio of 2 and up to tip speeds of 1000 feet per second. A cross section of the turbine rig is shown in figure 27. The rotor blades are integrally machined with an annular ring, which is attached to the rotor by means of mating circles of teeth cut into each side of the rotor and the ring. Through bolts (not shown) hold the ring and the disk together. The diameter of the three largest holes is 0.12 inch, that of the fourth is 0.09 inch, and that of the fifth is 0.07 inch. The coolant passages are not interconnected. The coolant is introduced as shown in figure 27 and the centrifugal force and the variation in the density of the cooled and heated liquid permit a constant circulation of the coolant within the blind holes. The coolant leaves the rotor through 12 axial-discharge holes and enters an annular collector. As in the aluminum turbine, the outside ends of the coolant holes are sealed with screwed-in plugs. No provision has been made as yet to recirculate the coolant. An individual coolant supply is provided for cooling the nozzle blades each of which has six drilled passages.

The turbine has been operated approximately 150 hours at inlet-gas temperatures up to 1910° R and tip speeds of 850 feet per second. The mass flows were up to two-thirds of design specifications. The rotor coolant flow was varied between 6 and 18 percent of the gas flow through the turbine. Unfortunately, the excessive nozzle-blade trailing-edge temperature prevented turbine operation at higher inlet-gas temperatures. Apparently the method of supplying coolant to the nozzle blades and the smallness (0.06 diam) of the trailing-edge hole, as well as its distance from the blade trailing edge, all combined to prevent sufficient cooling of this portion of the nozzle blade. Although the maximum coolant flow available (10 gal/min) was supplied to the nozzles, the trailing-edge temperature continued to read approximately twice as high as any other nozzle-blade temperature location. Thermocouples installed on the rotor blades indicated a maximum temperature of 1200° R along the leading edge at the blade tip at the maximum inlet-gas-temperature run of 1910° R with a 12-percent coolant flow. Preliminary data seem to indicate that the natural-convection coolant flow tends to break down at a certain point; further operation, however, at higher gas temperatures are required before any positive statement can be made.

Liquid-cooled 26-inch turbine aircraft-power-plant component. - This turbine is liquid-cooled by the natural-convection principle. It is designed for operation as a component of an aircraft power plant in conjunction with an axial-flow compressor. Unlike the liquid-cooled turbines previously described, the blades are not integrally machined with the rotor disks. The blades (2-in. span) are shown in figure 28. They are attached to the disk by means of a fir-tree arrangement. A sealed cavity in the base of each blade connects with five blind holes in each blade. A so-called coolant disk bolts to the rotor. Two passages (water in and water out) are provided in the coolant disk for each of the 80 reaction blades to permit coolant circulation.

The turbine was designed to produce 7980 horsepower at sea-level conditions with an inlet-gas temperature of 2700° R, an inlet pressure of 43 pounds per square inch absolute, a pressure ratio of 2.57, a mass flow of 42.5 pounds per second, and a tip speed of 1130 feet per second. At these operating conditions, a coolant flow through the rotor of 33 gallons per minute has been calculated as necessary to maintain a coolant temperature change of 560° R and a blade temperature of 1660° R.

Turbine Performance Characteristics

In evaluating the performance of an uncooled turbine, the ideal or isentropic power available in the gas is of primary concern. This thermodynamic power is used as a basis for computing turbine efficiency. From a knowledge of nozzle, blading, and tip-leakage losses, evaluation of the power actually delivered to the blades is possible. Subtraction of the disk-friction loss and the bearing and gear losses from the blade power determines the shaft power of the machine.

The procedures for evaluating and correlating these performance factors for an uncooled turbine are well established, but in the cooled turbine, no established basis exists for determining the thermodynamic performance in the presence of heat transfer and with two separate fluids moving through the turbine. In the air-cooled turbine, the situation is further complicated by the mixing of the coolant stream with the working fluid, which results in entropy increases. The effects of heat transfer tend to mask some of the aerodynamic losses and separation of the various losses occurring in the two streams is difficult. This separation requires new definitions of efficiencies and introduces new parameters into correlation of experimental data. For blade power, the heat transfer is introduced into the energy equation and the internal-coolant pumping power must be included in evaluating either the shaft power or the torque at the blade root. For the cooled turbine, possible parameters for correlation of pumping power, blade power, and efficiency are

$$\frac{P_p \dfrac{w_g}{w_a}}{\delta_1 \sqrt{\theta_1}} = f\left(\frac{N}{\sqrt{\theta_1}}, \frac{p'_{m,3}}{p'_{g,1}}\right)$$

$$\frac{P_B}{\delta_1 \sqrt{\theta_1}} = f'\left(\frac{p'_{m,3}}{p'_{g,1}},\ \frac{N}{\sqrt{\theta_1}},\ Re_g,\ \frac{Q_a}{\delta_1 \sqrt{\theta_1}}\right)$$

$$\eta_{ad} = \frac{(w_a\, h'_{g,1} + w_a\, h'_{a,h}) - (w_g + w_a)\, h'_{m,3} - Q_S}{w_g\, h'_{g,1}\left[1 - \left(\frac{p'_{m,3}}{p'_{g,1}}\right)^{\frac{\gamma_g - 1}{\gamma_g}}\right] w_a\, h'_{a,h}\left[1 - \left(\frac{p'_{m,3}}{p'_{a,h}}\right)^{\frac{\gamma_g - 1}{\gamma_g}}\right]}$$

No investigations have yet been made to verify these parameters or to determine the functions that relate them. Analytical methods, however, are being developed to assist those working in the turbine-cooling field and to provide suggested methods for obtaining and reducing experimental data.

Engine Performance Characteristics

The final measure of the effectiveness of turbine cooling, either for nonstrategic blades or for high-temperature machines, is in the performance of the engine. This type of evaluation requires many additional methods of analysis to determine the performance and the losses of a cooled engine from experimental data on the same engine uncooled or to predict the performance of an engine in the design stage. The general procedure is to apply the best available heat-transfer data for the outside and the inside of the cooled blade and to determine the coolant-flow requirements for the range of operating conditions imposed on the turbine. These operating conditions are functions of turbine configuration, turbine-inlet temperature, turbine speed, and compressor pressure ratio, as well as flight Mach number and altitude of the aircraft. Ranges of altitude must be covered because of the significant effects on coolant requirements and losses. In an air-cooled engine, the principal factors considered are the thermodynamic effects of heat loss, friction and compression in blade passages and disk, the internal pumping power in the disk and the blades, the external pumping power in the compressor before bleedoff, the decrease in mass flow through the turbine, and the mixing of the coolant with the working fluid. The summation of these effects in a turbojet engine is a decrease in jet-nozzle inlet-total temperature and pressure, which adversely affects performance of the engine.

Several effects of cooling on the performance of gas-turbine stages have been qualitatively investigated in reference 38. In this analysis, the flow of the combustion gas through the turbine stage was approximated by flow through a tapered conical tube. Three phases of the effect of cooling on gas-turbine-stage performance are significant: (1) The cooling requirements of the stage must be minimized; (2) the decrease in stage efficiency due to cooling must be minimized; and (3) the later stages in the turbine must be permitted only a minimum reduction in available work.

According to reference 38, the cooling requirements of the stage can be kept at a minimum by employing water cooling only in the rotor and by cooling the less highly stressed stator by means of air. As previously mentioned, porous-blade cooling is probably adequate for the cooling of the stator and, in addition, such a type of cooling increases the mass flow through the turbine. Cooling requirements may be minimized in the water-cooled rotor either by fabricating the blades from a material with a high thermal conductivity to expedite the flow of heat from the blades and into the coolant or by coating the blades with a very low-conductivity material so as to decrease the heat flow from the hot gas to the blades. The analysis indicates that high blade speeds should be employed so as to increase the work output per stage. A high work output per stage decreases the required number of stages and, consequently, the cooling requirements of the turbine.

The reduction of the available work in the later stages of the gas turbine is outside the scope of reference 38, but, as mentioned in that reference, it constitutes an effect that in many cases is more significant than the slight decrease in stage efficiency due to the cooling of the stage.

CONCLUDING REMARKS

Some idea concerning the gains to be obtained by the use of high turbine-inlet temperature, the possibilities of achieving these gains by using turbine cooling, and the problems involved in such research have been presented. The potentialities are so great and the need so urgent for cooling of turbine blades, especially those made of metals that have low strategic-alloy content, that a tremendous amount of effort put into the over-all problem is warranted on the basis of analyses and experiments made to the present time.

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Cleveland, Ohio.

APPENDIX - SYMBOLS

The following symbols are used in this report:

A	cross-sectional area, sq ft
B	force, lb
b	length of blade (span), ft
C	length of blade chord, ft
c_p	specific heat at constant pressure, Btu/(lb)(°F)
D	characteristic dimension in Reynolds and Nusselt numbers, ft
f	friction coefficient, also function of
Gr	Grashof number, $\frac{gD^3 \beta_f \rho_f^2 (T_B - T_f)}{\mu_f^2}$
g	ratio of absolute to gravitational unit of mass, lb/slug or acceleration due to gravity, ft/sec^2
H	convection heat-transfer coefficient, Btu/(sec)(sq ft)(°F)
h'	enthalpy based on total temperature, Btu/lb
J	mechanical equivalent of heat, 778.3 ft-lb/Btu
k	thermal conductivity, Btu/(°F)(ft)(sec)
L	length, ft
l	perimeter, ft
M	Mach number
N	speed of rotation, rpm
Nu	Nusselt number, $\frac{HD}{k_f}$
P	power, hp

p'	total pressure, lb/sq ft
Pr	Prandtl number, $\frac{c_{p,f}\,\mu_f\,g}{k_f}$
Q	heat flow rate, Btu/sec
Re	Reynolds number, $\frac{\rho_f D W_f}{\mu_f}$
r	radius, ft; (in radiation equation, distance between two areas being considered, ft)
s	stress, lb/sq in.
T	static temperature, °R
T'	total temperature, °R
T''	total temperature relative to moving blade, °R
V	turbine tip speed, ft/sec; absolute fluid velocity, ft/sec
W	relative velocity, ft/sec
w	weight flow, lb/sec
X	$\frac{\omega^2 r_h w_a}{gJH_o l_o\left[T_{g,e} - (T_{a,e})_h\right]}$
x	radial distance from blade root to point on blade span considered, ft
Y	$\frac{\omega^2 w_a^2(\lambda + 1)\, c_{p,a}}{gJH_o^2 l_o^2\left[T_{g,e} - (T_{a,e})_h\right]}$
y	distance normal to wall, ft

Z $\quad \frac{1}{\lambda + 1} \frac{H_o l_o b}{w_a c_{p,a}} \frac{x}{b}$

α surface absorptivity

β coefficient of thermal expansion, $1/^{\circ}R$

δ $\quad \frac{p_{g,1}}{\text{NACA standard sea-level pressure}}$

η efficiency

θ $\quad \frac{T_{g,1}}{\text{NACA standard sea-level temperature}}$ (when used with radiation equation (15) it is the angle between the normal to one area and a line joining the two areas, deg)

Λ recovery factor

λ $\quad \frac{H_o l_o}{H_i l_i}$

μ absolute viscosity, slugs/(sec)(ft)

ρ density, slugs/cu ft

σ Stefan-Boltzmann constant, Btu/(sec)(sq ft)($^{\circ}R^4$)

τ blade spacing/blade chord

ω angular velocity, radians/sec

ψ turning angle, deg, or function

ψ', ψ'', ψ''' function

Subscripts:

a cooling air

ad adiabatic

av	average
B	blade
c	compressor, critical point
e	effective (used with symbol for temperature and denotes temperature effecting heat transfer)
f	fluid
g	combustion gas
h	blade root
i	inside surface of blade
m	mixture (refers to mixing of combustion gases and cooling air)
o	outside surface of blade
P	pumping
R	rotor blades
r	radiation
S	stator blades
s	refers to isentropic process
w	wall
x	point along blade span
max	maximum
0	free stream
1	turbine inlet
2	turbine outlet
3	downstream of turbine where complete mixing of combustion gases and cooling air occur

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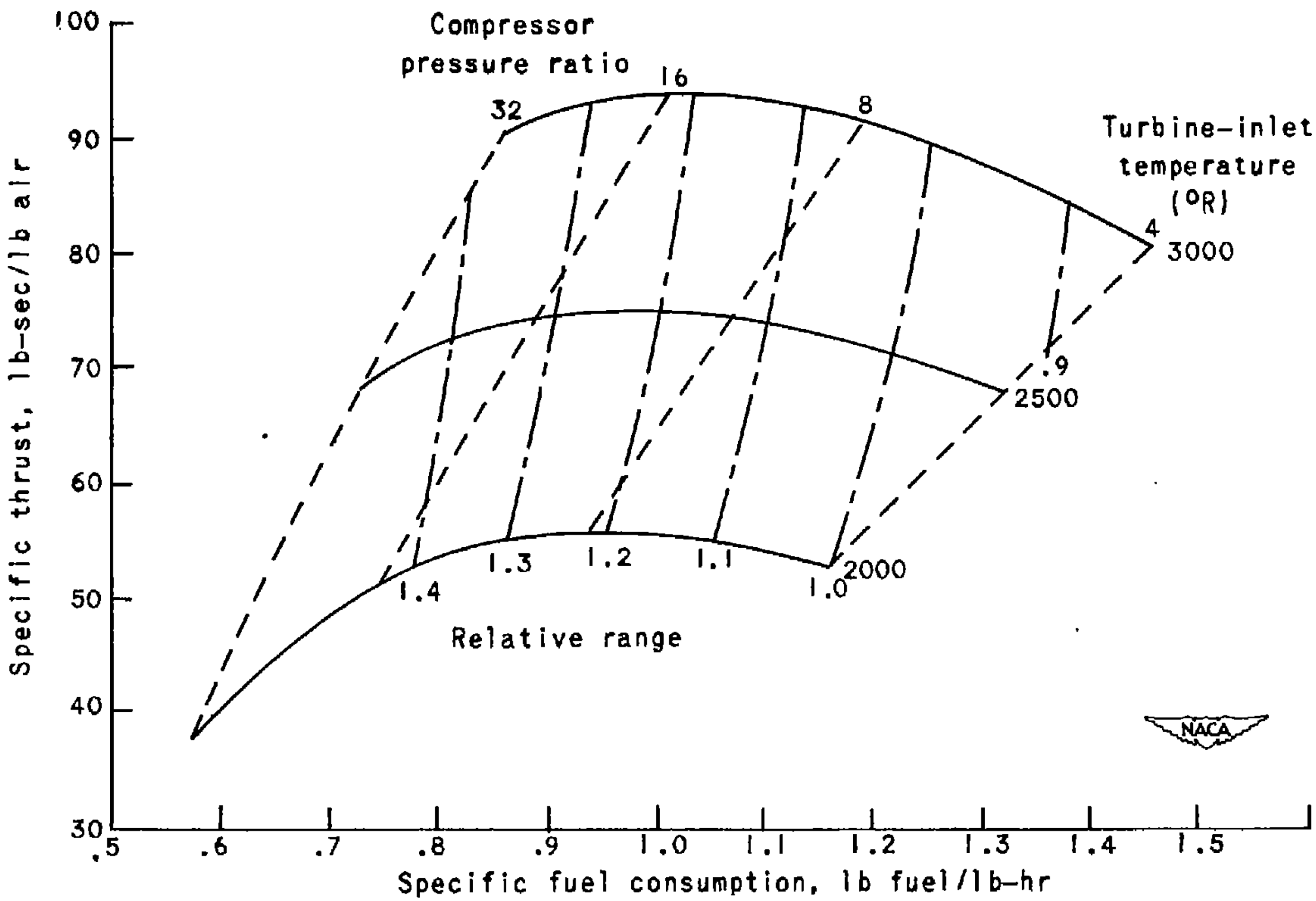
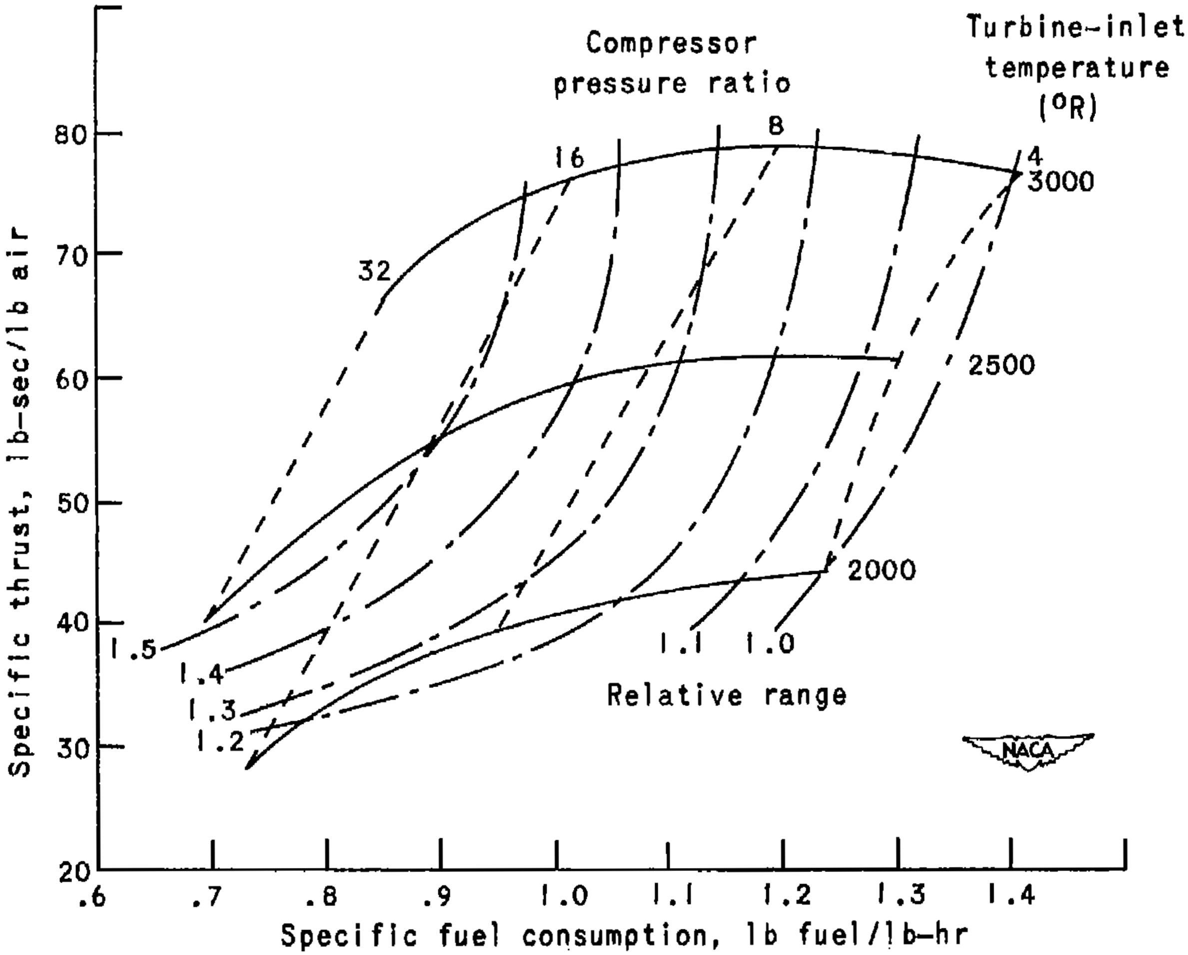


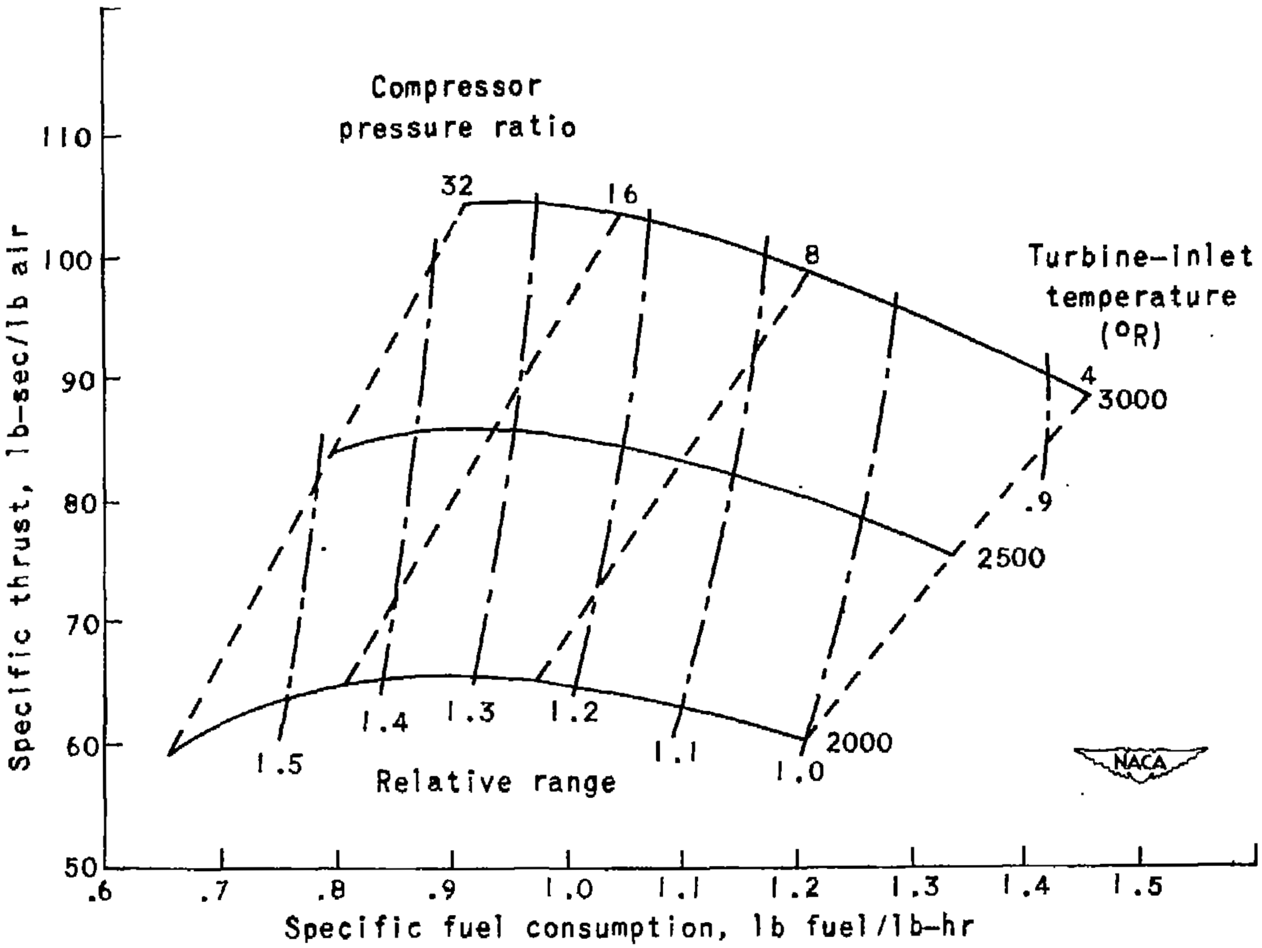
(a) Airspeed, 500 miles per hour; altitude, sea level.

Figure 1. - Turbojet-engine performance. Compressor and turbine efficiencies, 0.90.



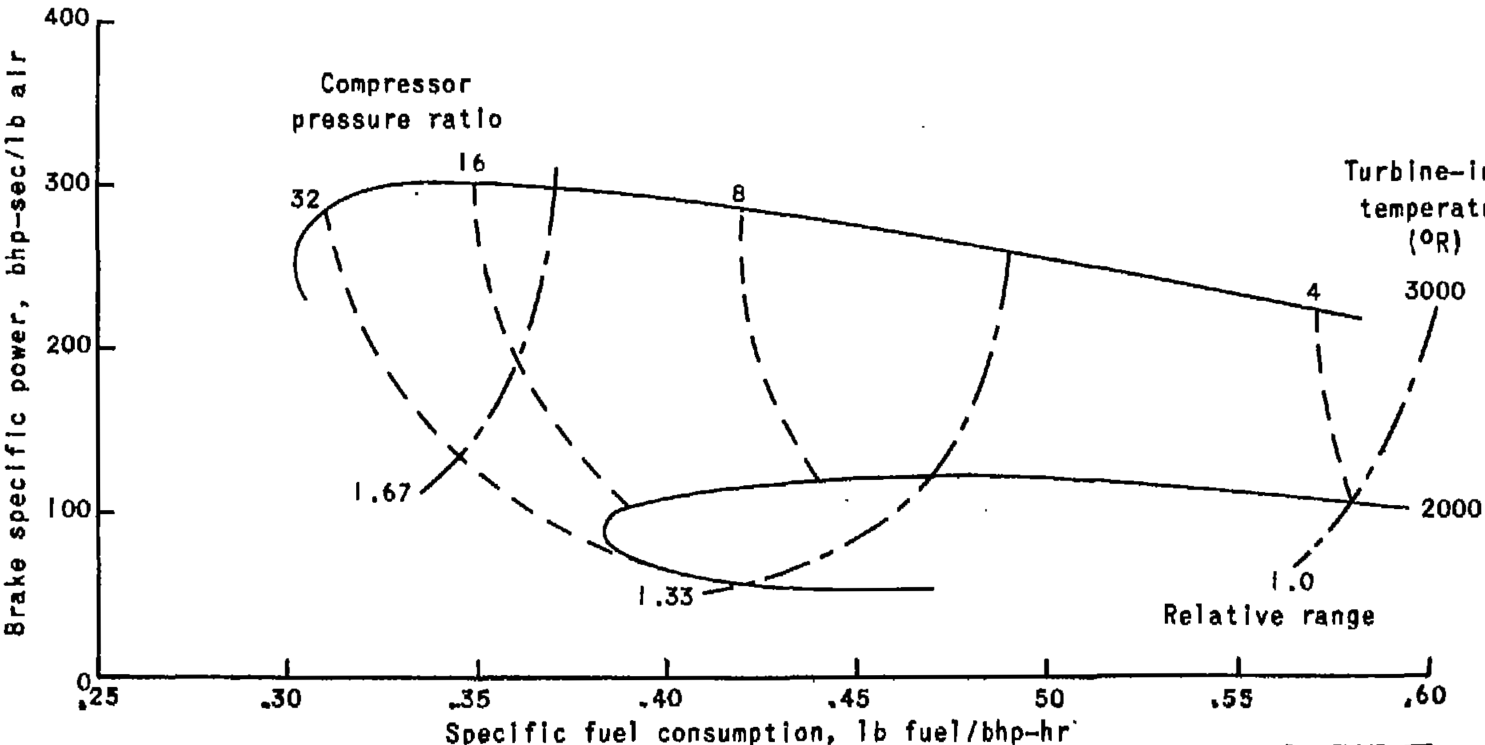
(b) Airspeed, 1000 miles per hour; altitude, sea level.

Figure 1. - Continued. Turbojet-engine performance. Compressor and turbine efficiencies, 0.90.



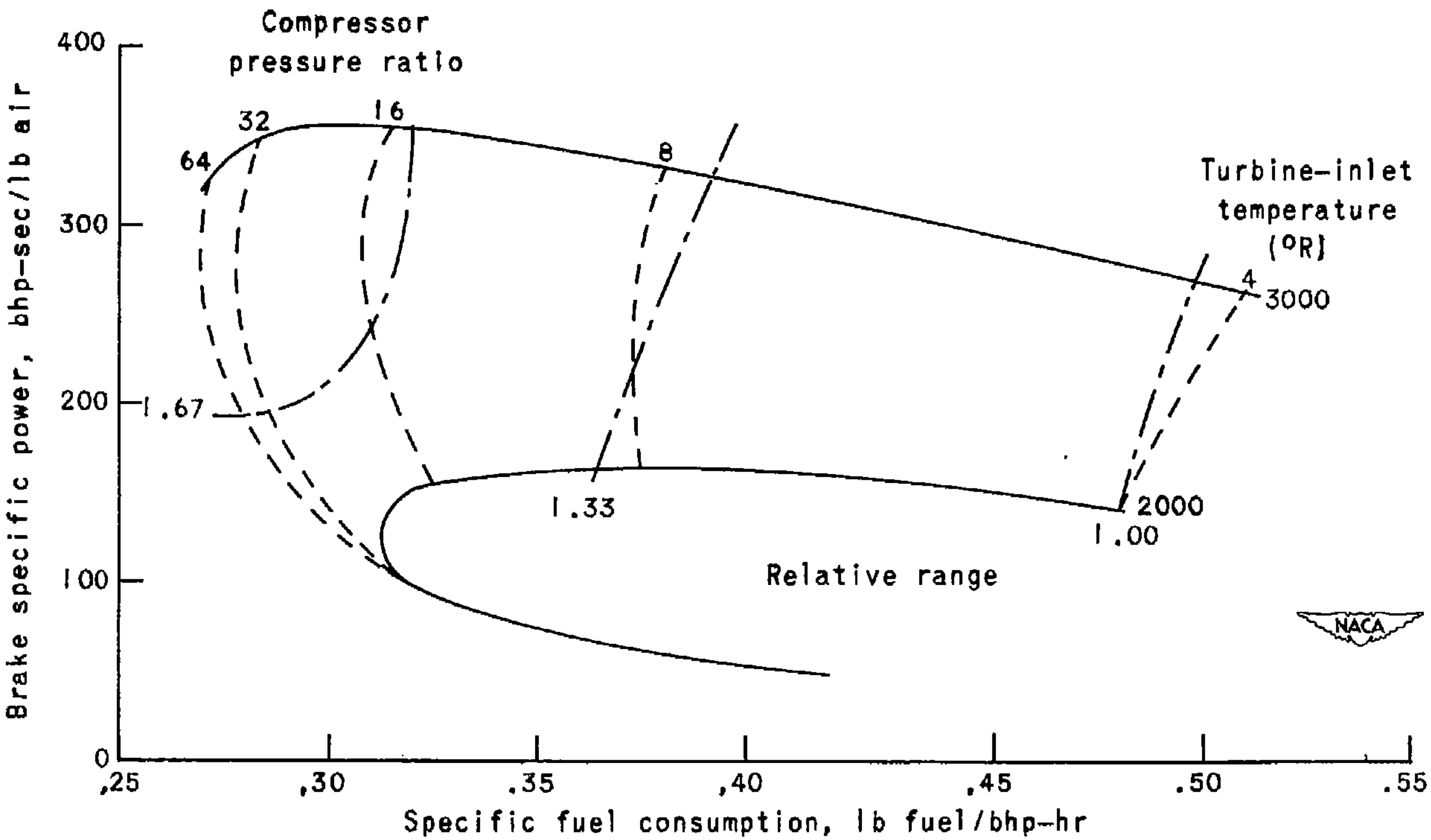
(c) Airspeed, 500 miles per hour; altitude, 30,000 feet.

Figure 1. - Concluded. Turbojet-engine performance. Compressor and turbine efficiencies, 0.90.



(a) Take-off conditions; altitude, sea level.

Figure 2. - Turbine-propeller-engine performance. Compressor and turbine efficiencies, 0.90; propeller efficiency, 0.80.



(b) Airspeed, 400 miles per hour; altitude, 30,000 feet.

Figure 2. - Concluded. Turbine-propeller-engine performance. Compressor and turbine efficiencies, 0.90; propeller efficiency, 0.80.

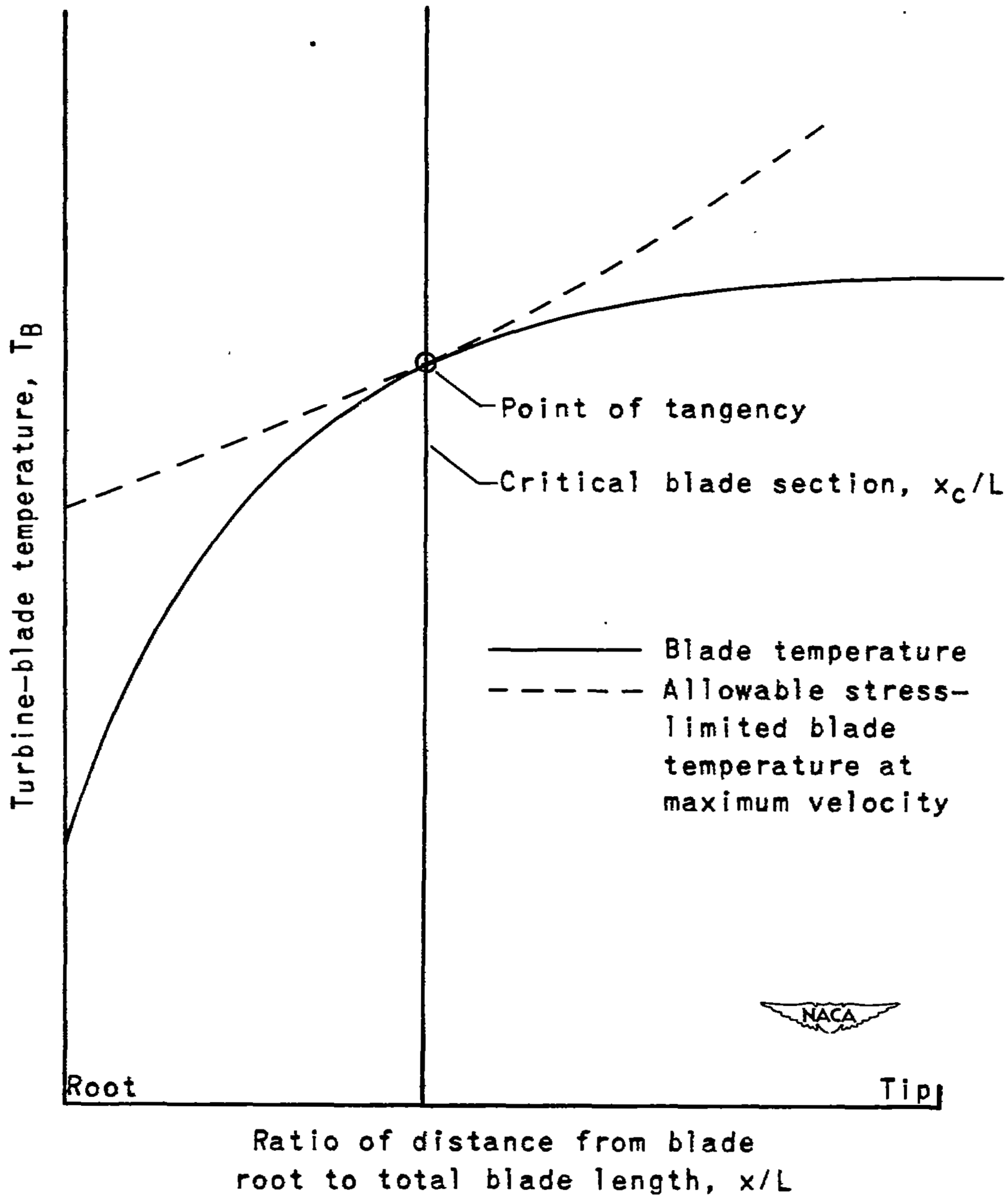


Figure 3. - Method of determining limiting dilution and critical blade section.

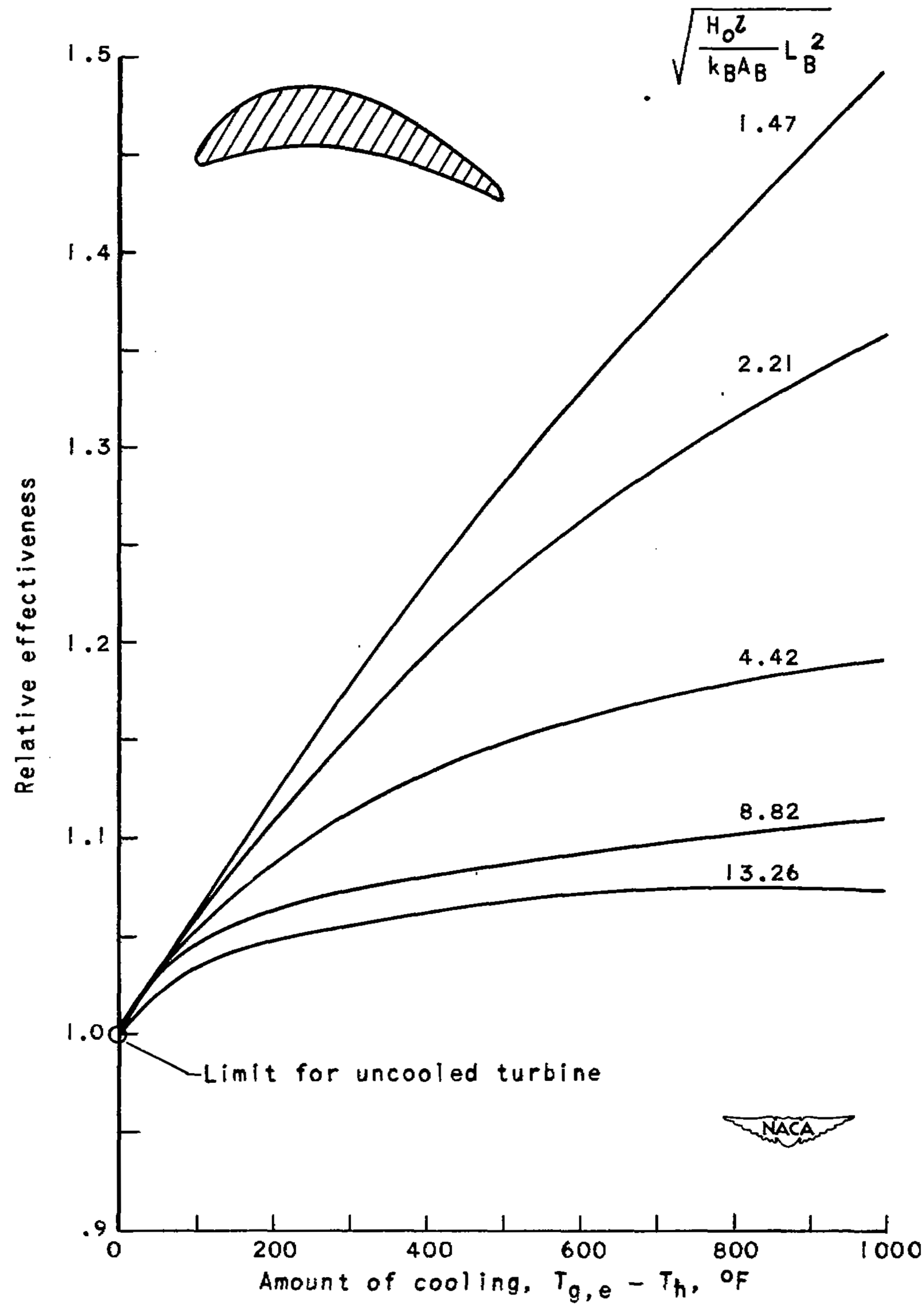


Figure 4. - Variation of rim-cooling effectiveness.

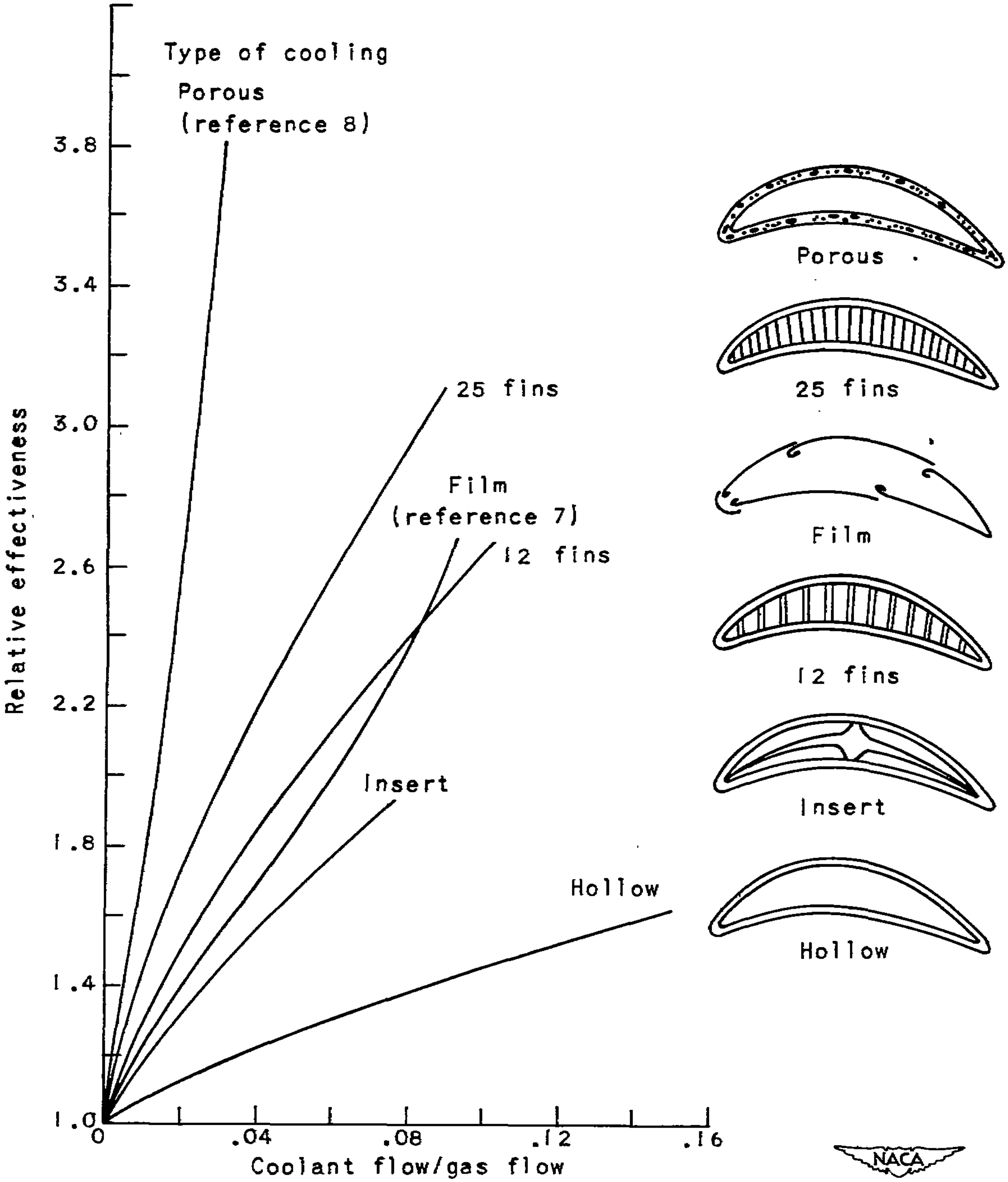


Figure 5. - Comparison of various types of air cooling for gas-turbine blades made of S-816.

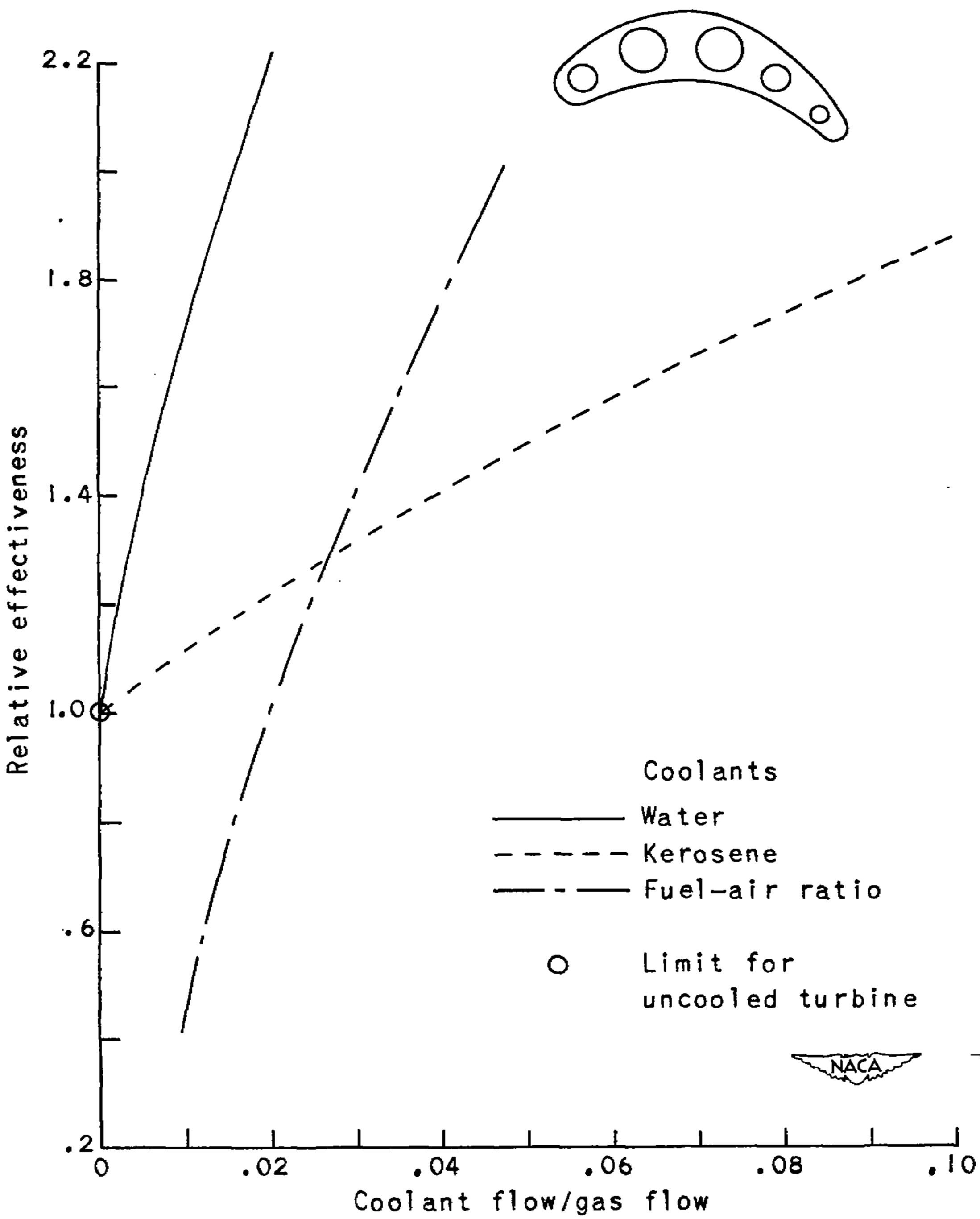


Figure 6. - Effectiveness of water and kerosene as coolants for gas-turbine blades made of S-816.

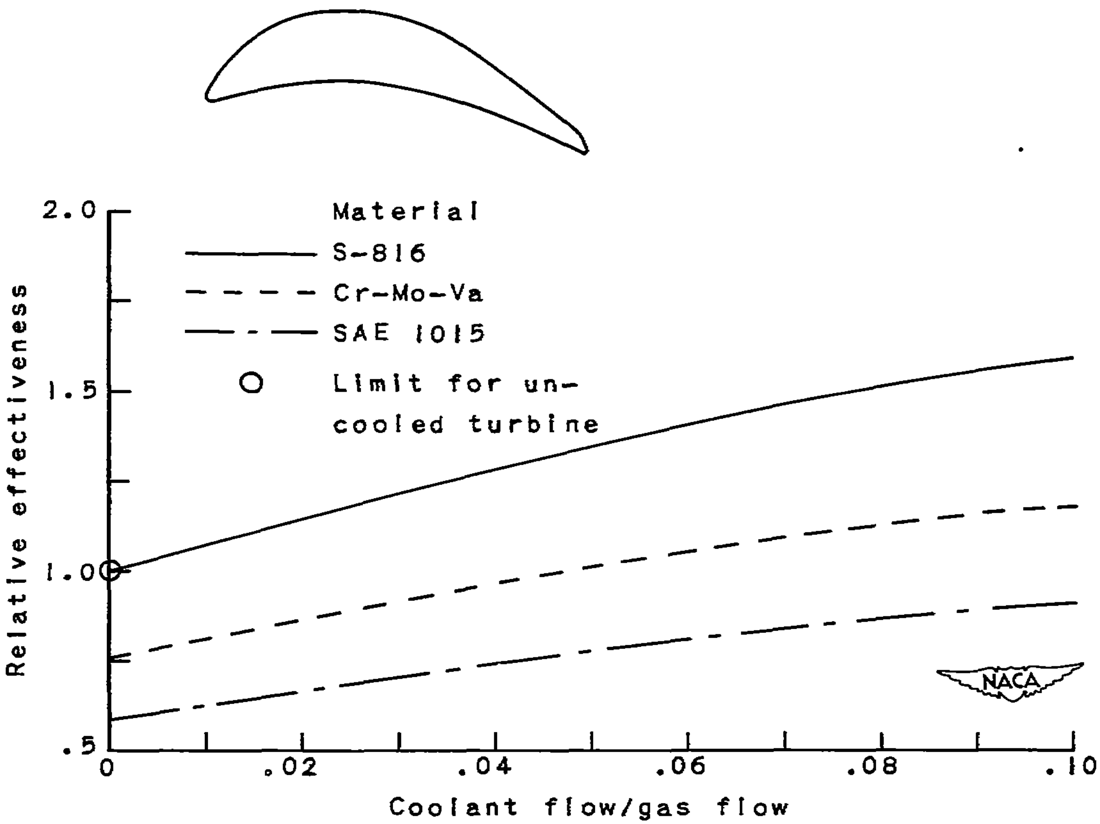


Figure 7. - Cooling-air requirements of simple hollow turbine blades made of various materials.

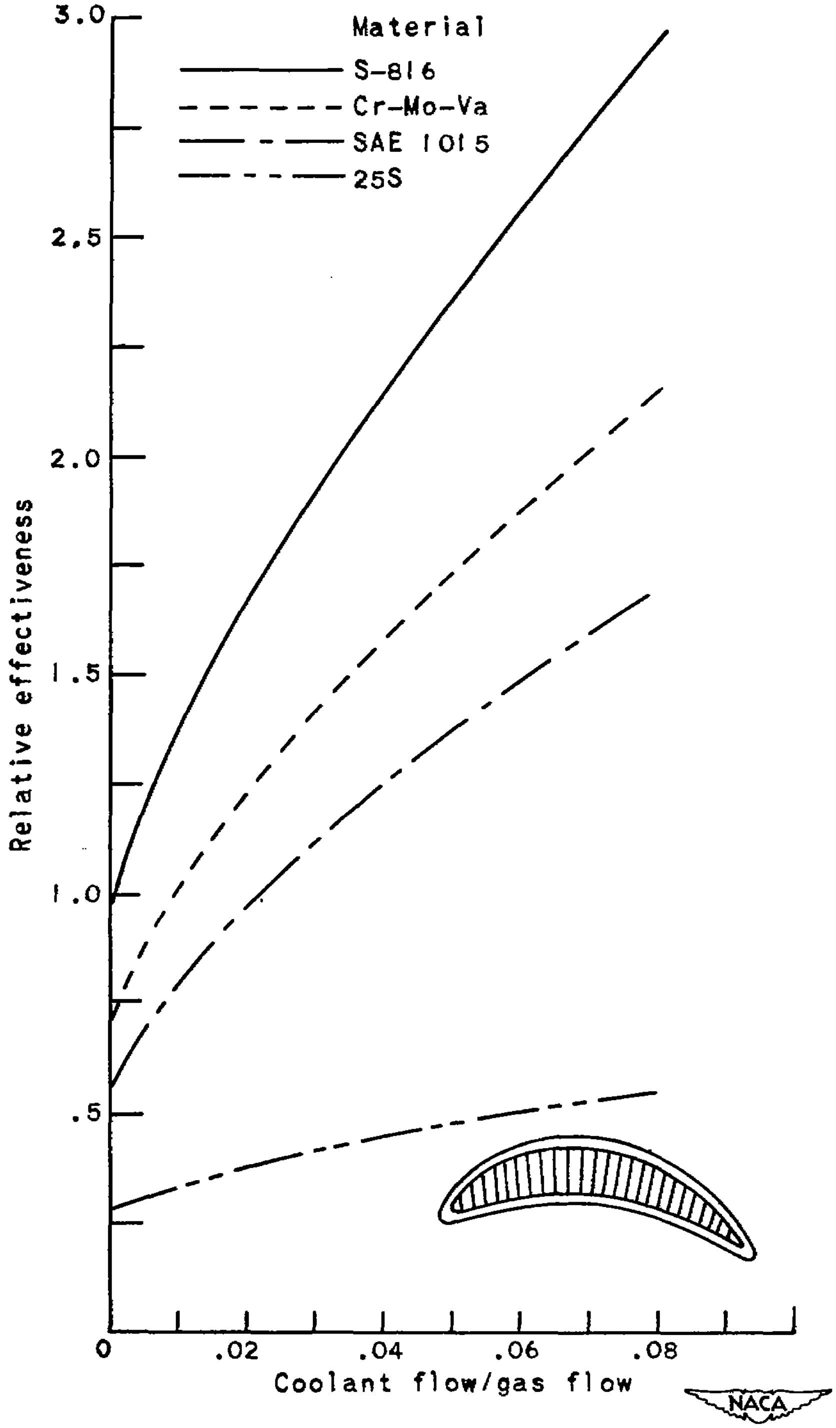


Figure 8. - Cooling-air requirements of 25-fin blades made of various materials.

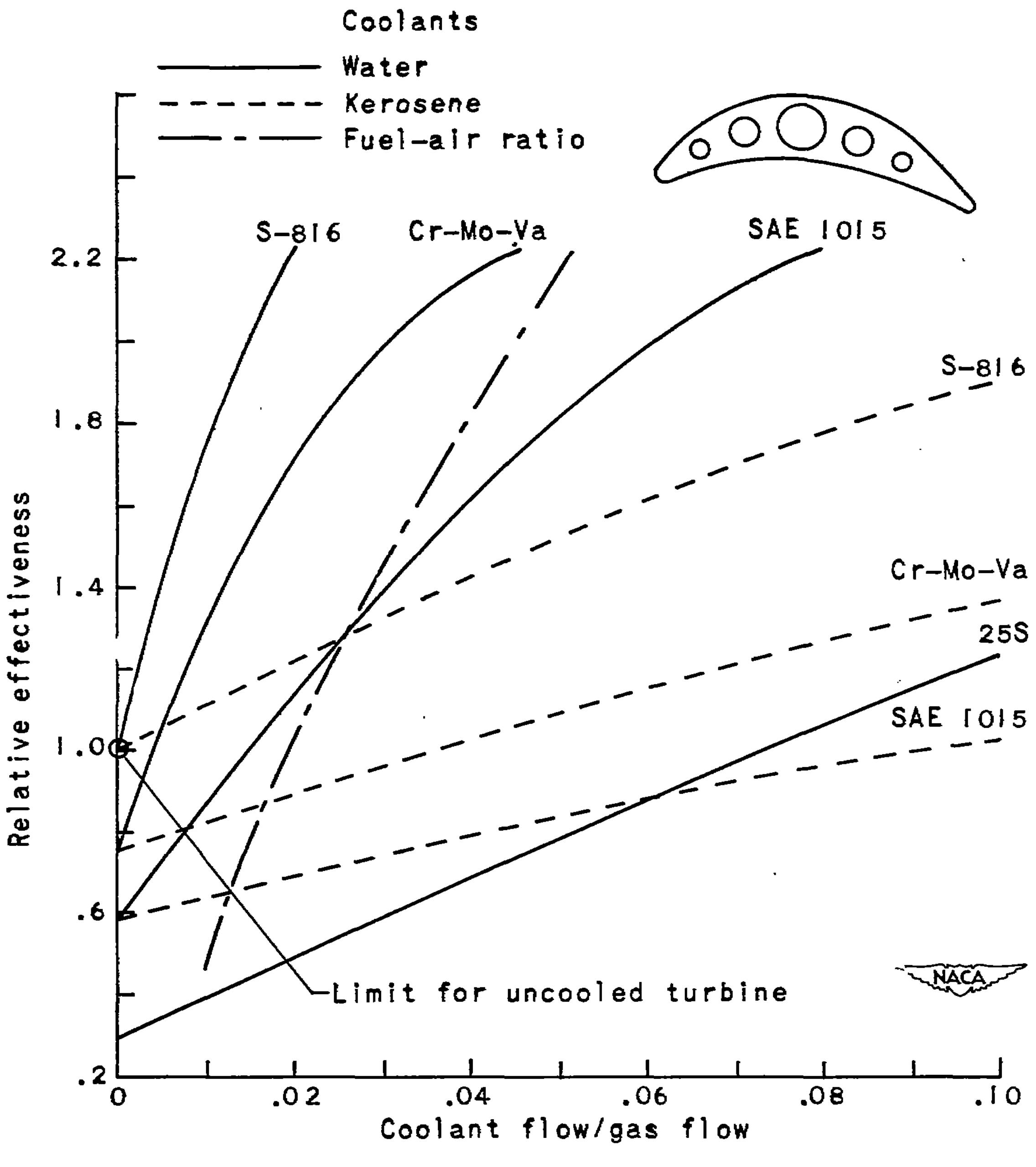


Figure 9. - Effectiveness of water and kerosene as coolants for gas-turbine blades made of various materials.

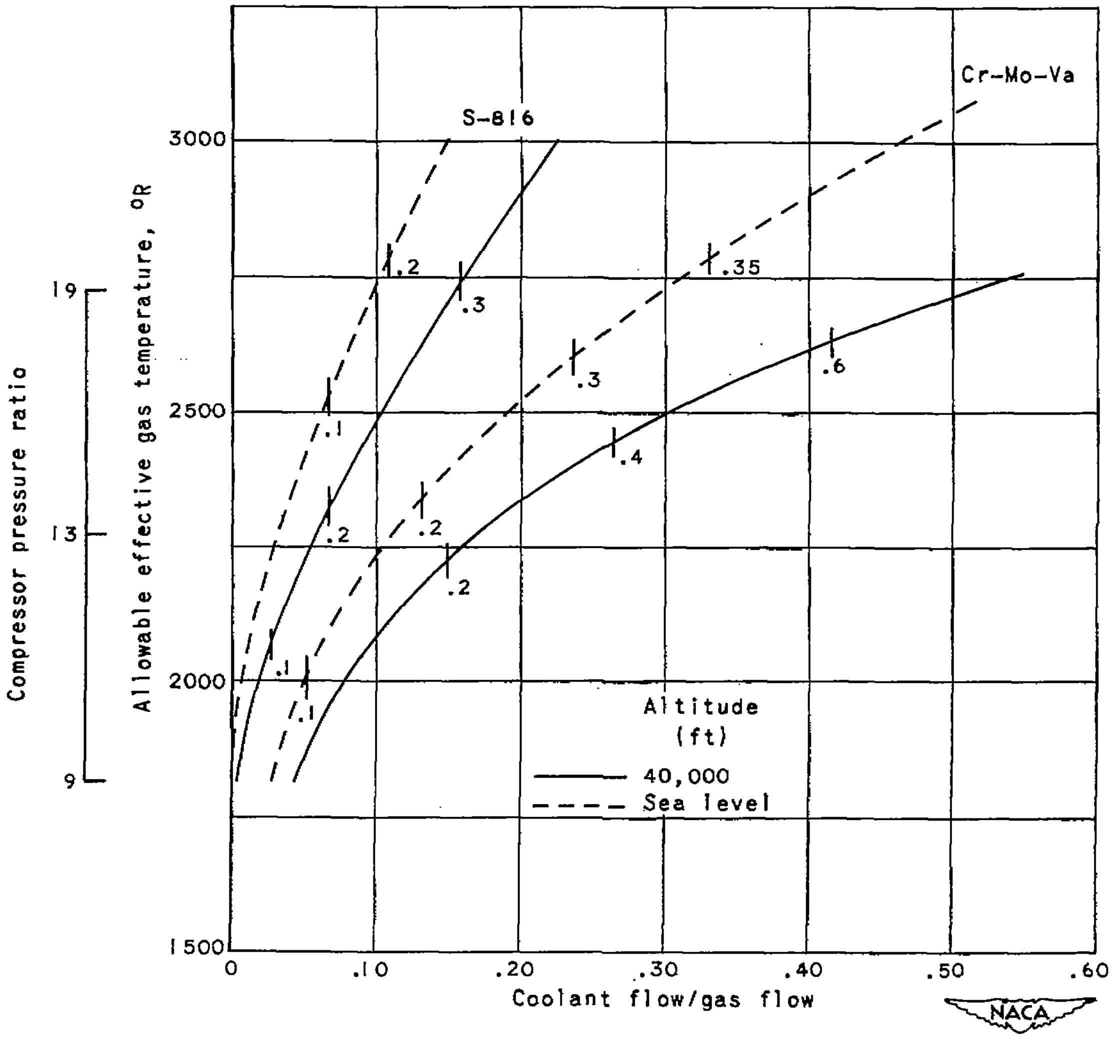


Figure 10. - Cooling-air requirements for first-stage turbine of turbojet-engine optimum-power pressure ratio. Flight speed, 500 miles per hour; compressor and turbine efficiencies, 0.85. Values beside vertical lines denote coolant Mach number at blade tip.

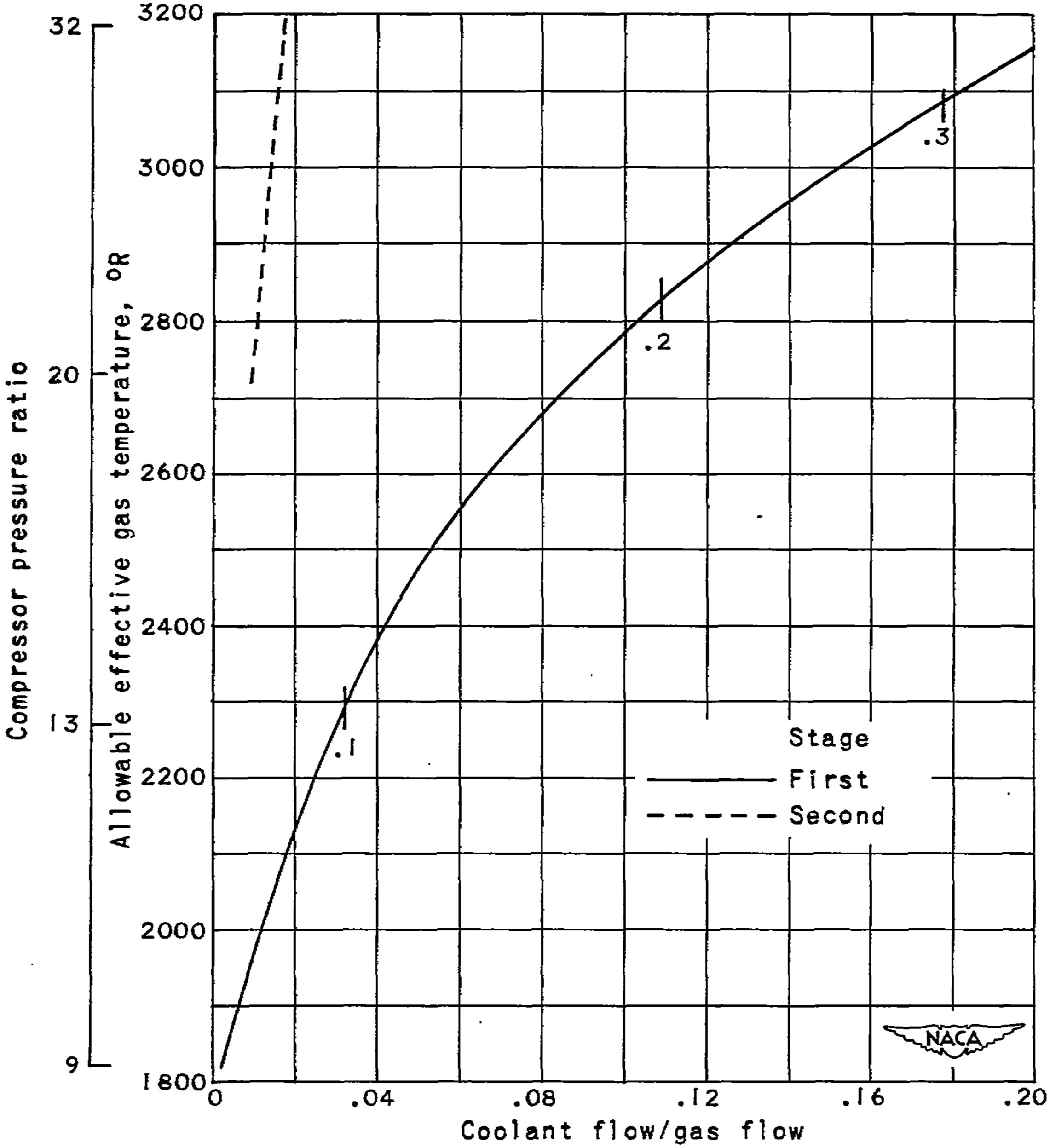
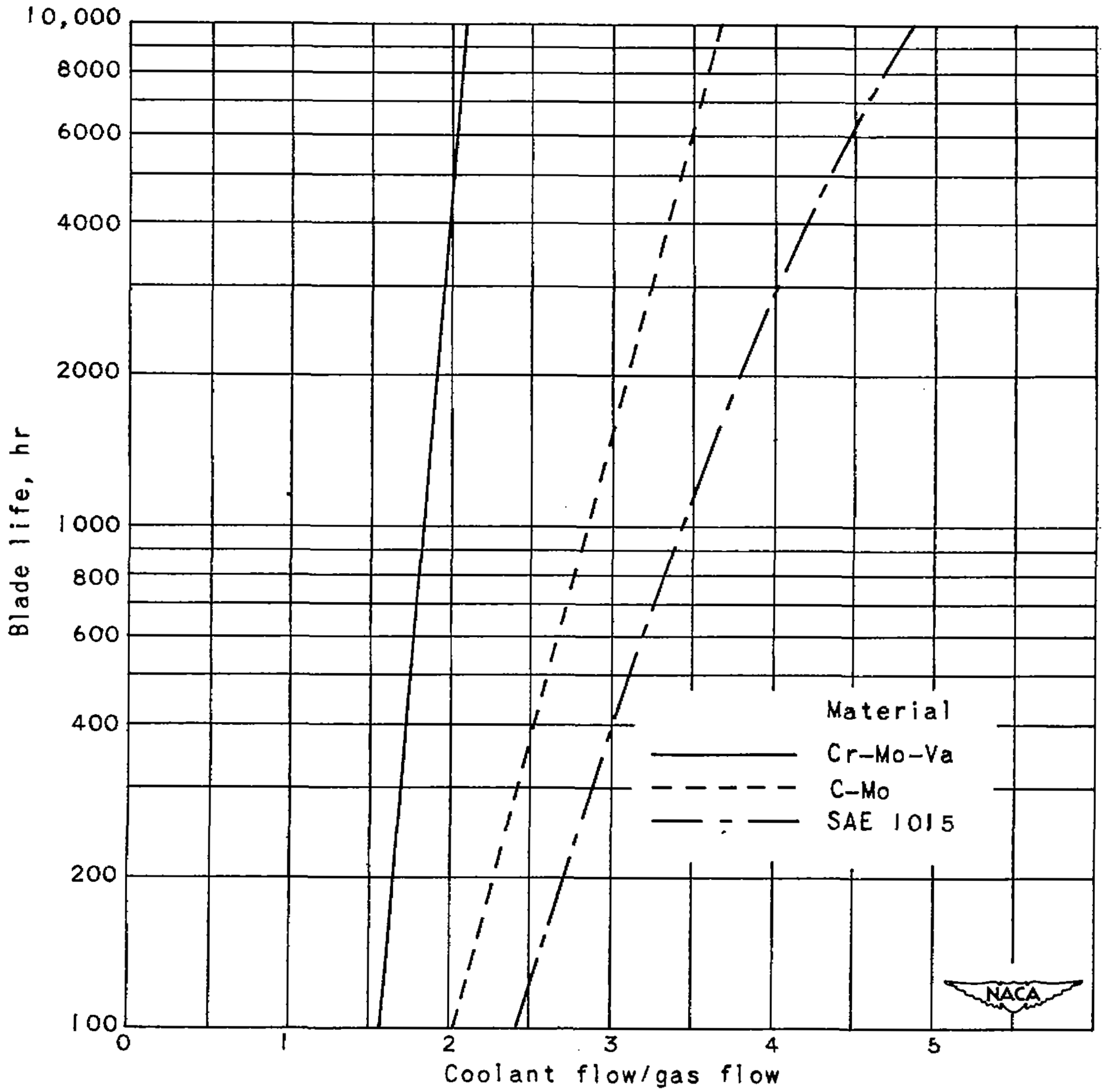
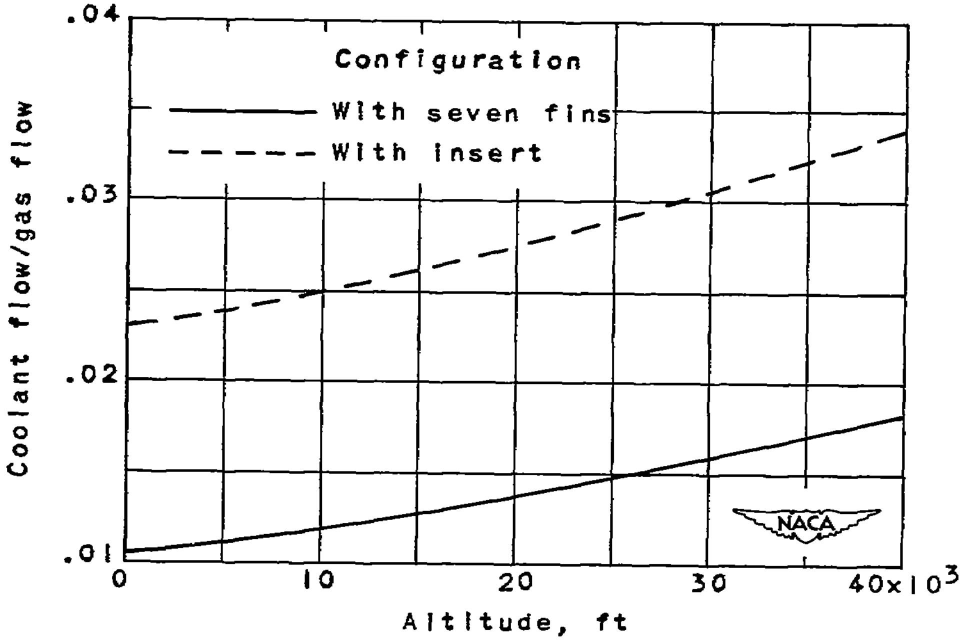


Figure 11. - Required cooling-air flow for turbine-propeller engine having 25-fin turbine blades made of S-816; optimum-power compressor pressure ratio; altitude, 35,000 feet. Values beside vertical lines denote coolant Mach number at blade tip.



(a) Variation of blade life with cooling-air flow for seven-fin blades made of various materials for typical turbine; altitude, 40,000 feet; turbine-inlet temperature, 1500° F.

Figure 12. - Variation of required cooling-air flow with blade life, blade material, coolant-passage configuration, and altitude.



(b) Variation of cooling-air flow required with altitude for two coolant-passage configurations for typical turbine; turbine-inlet temperature, 1500° F; material, Cr-Mo-Va; blade life, 1000 hours.

Figure 12. - Concluded. Variation of required cooling-air flow with blade life, blade material, coolant-passage configuration, and altitude.

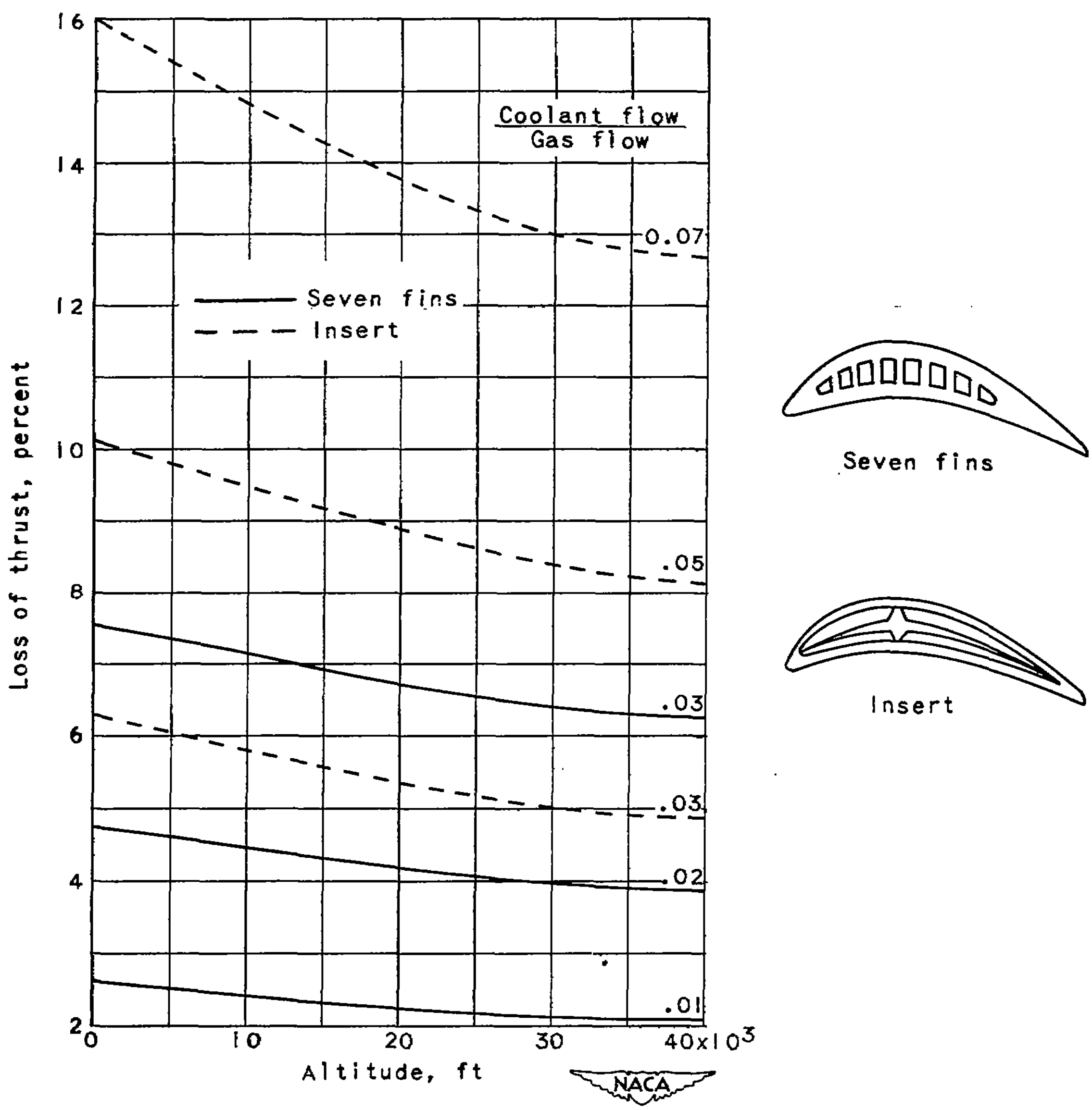


Figure 13. - Loss of thrust at various altitudes for three ratios of coolant flow to gas flow for typical turbojet engine. Flight Mach number, 0.8.

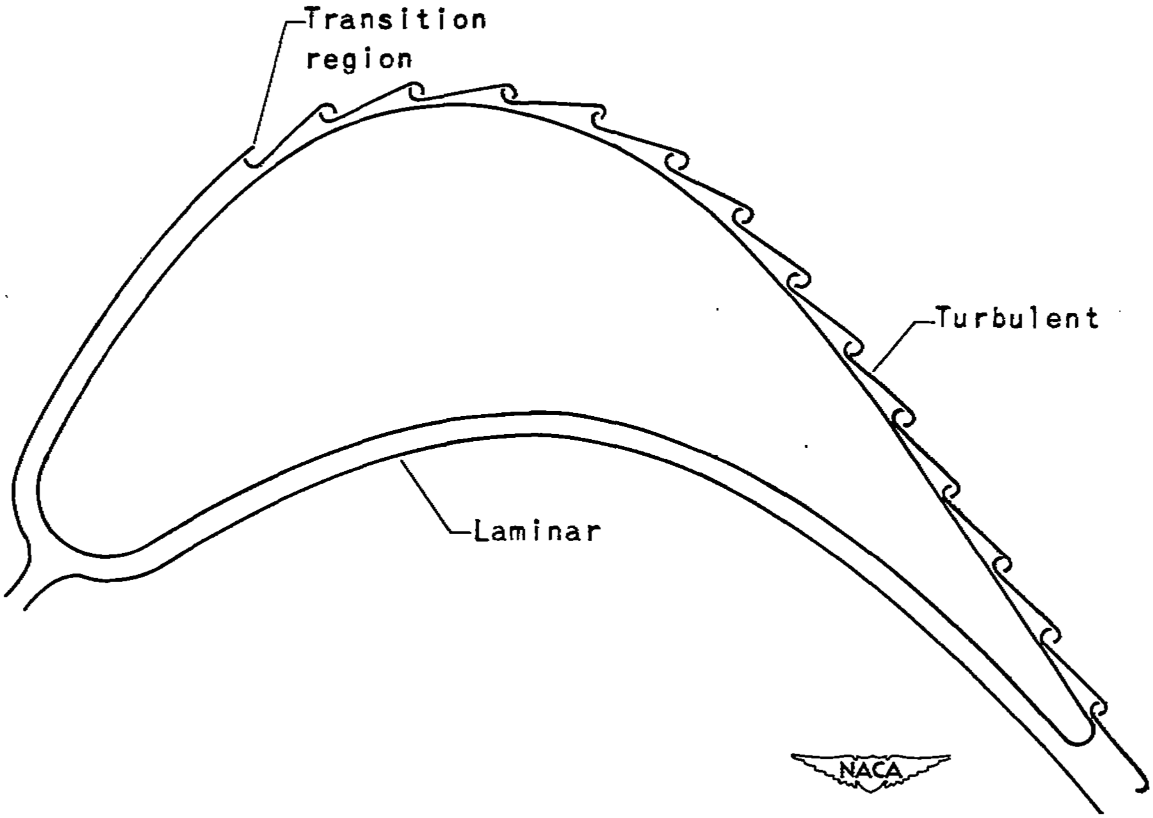
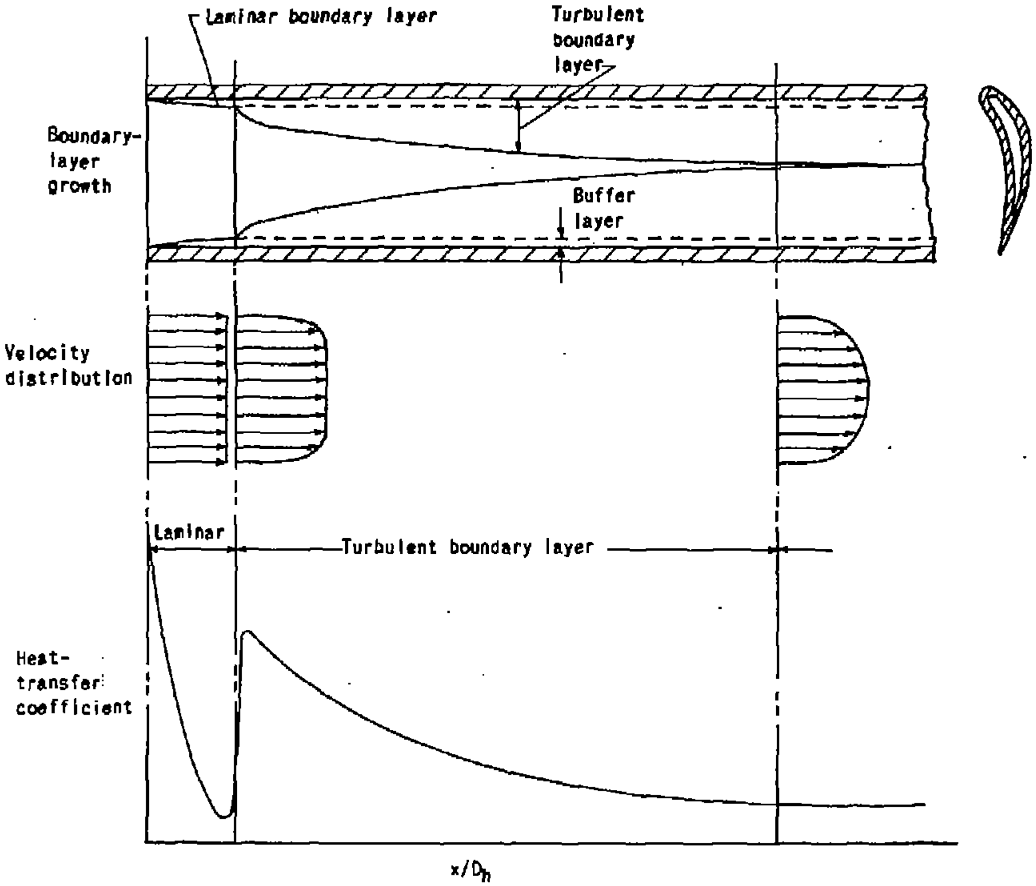
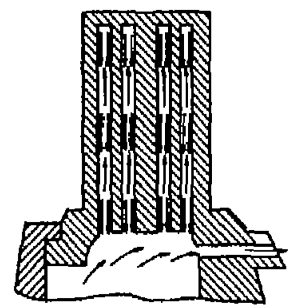


Figure 14. - Boundary-layer flow of turbine blade.



(a) Inlet effects on heat-transfer coefficients inside air-cooled turbine blades.



(b) Natural convection inside liquid-cooled blades.

Figure 15. - Characteristics of coolant flow in air-cooled and liquid-cooled turbine blades.

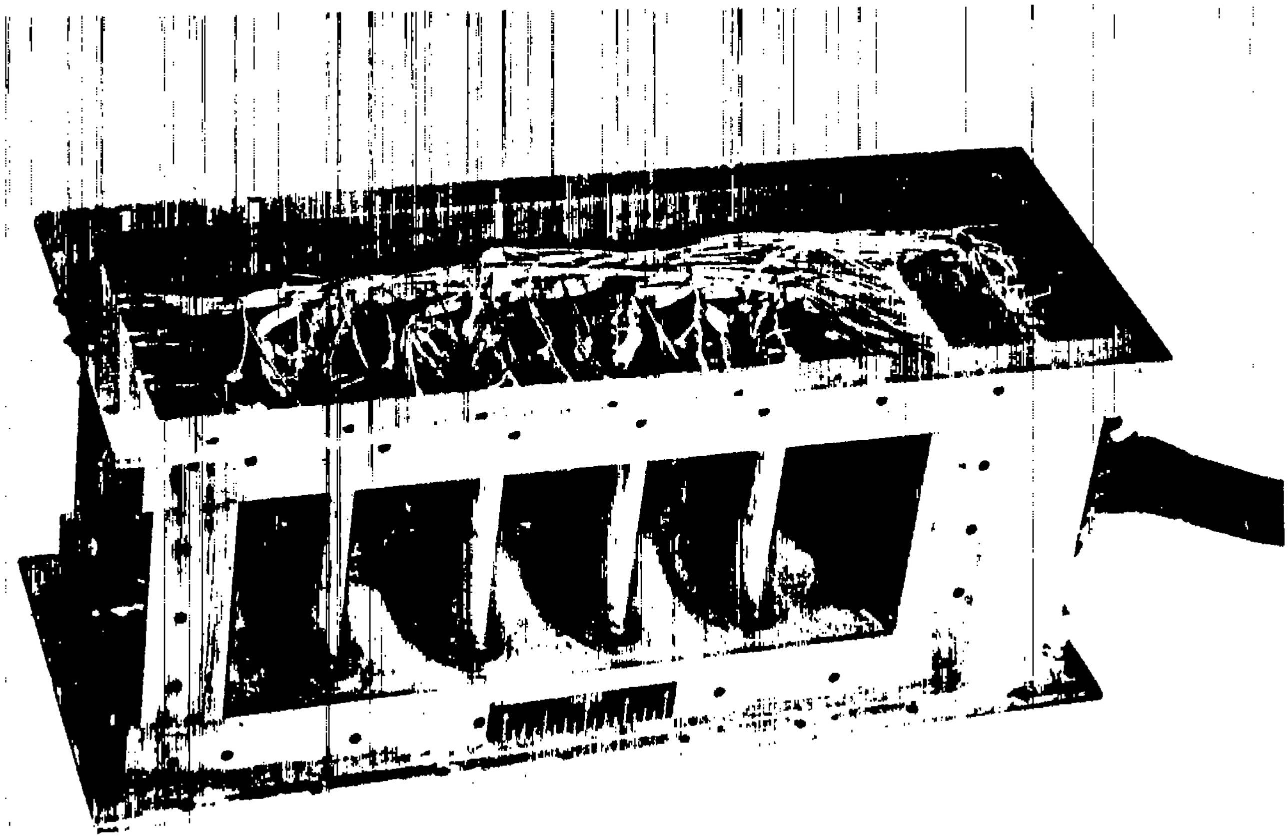


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Figure 16. - Typical air-cooled static cascade for determining local and average heat-transfer coefficients.

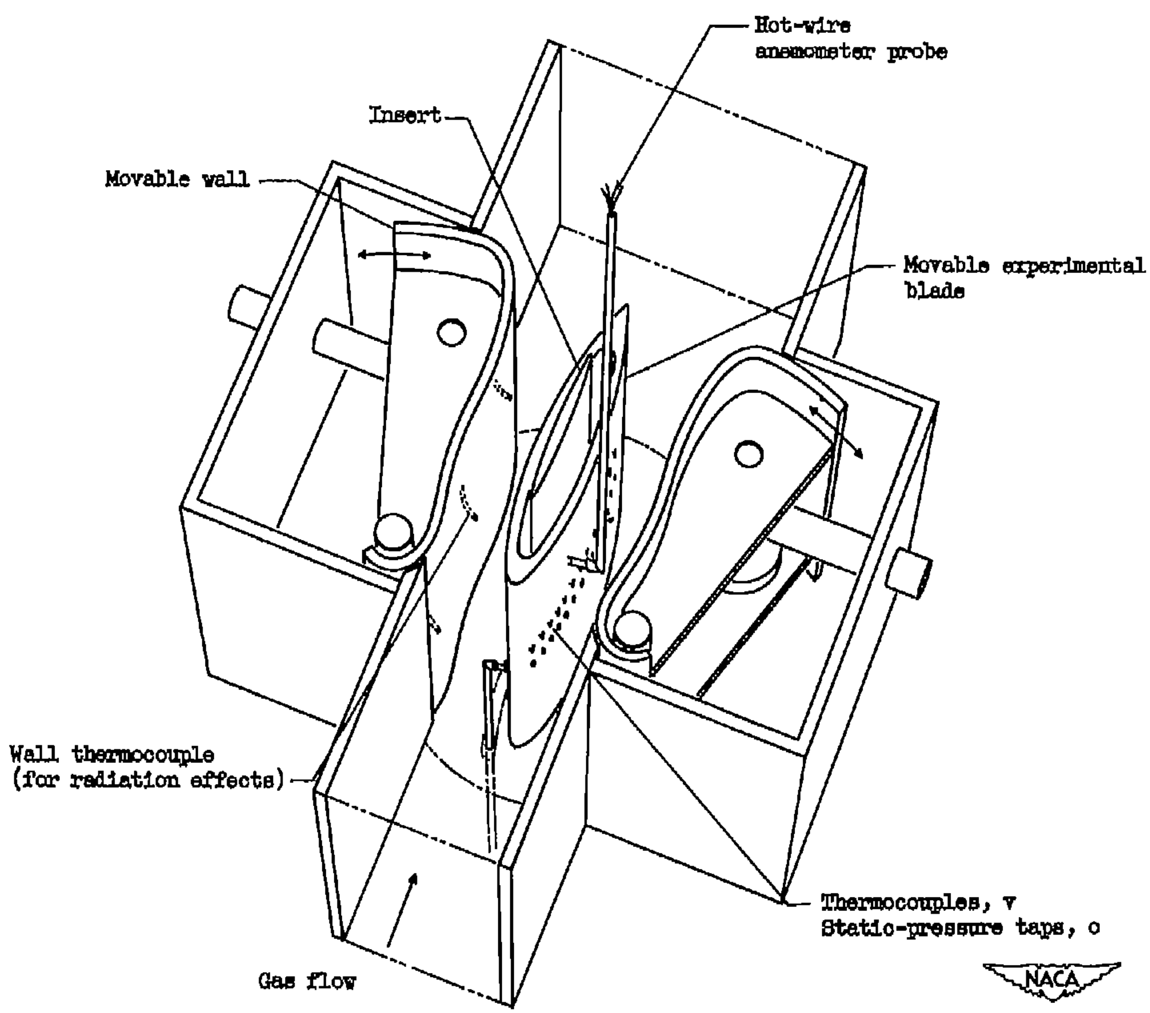


Figure 17. - Special turbine-cooling static boundary-layer rig.

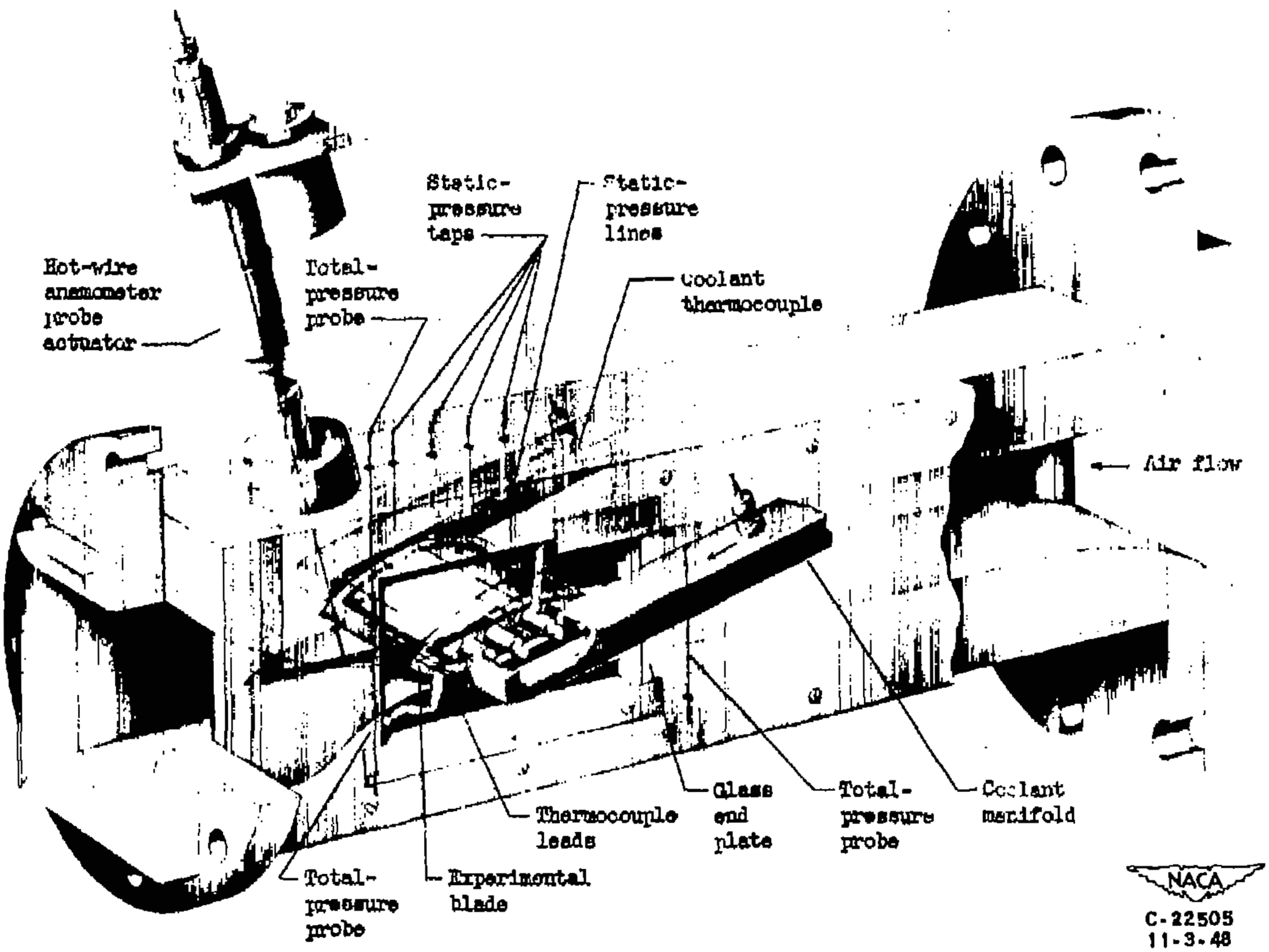


Figure 18. - Test section for investigation of boundary layer around cooled blade by means of Mach-Zehnder interferometer.

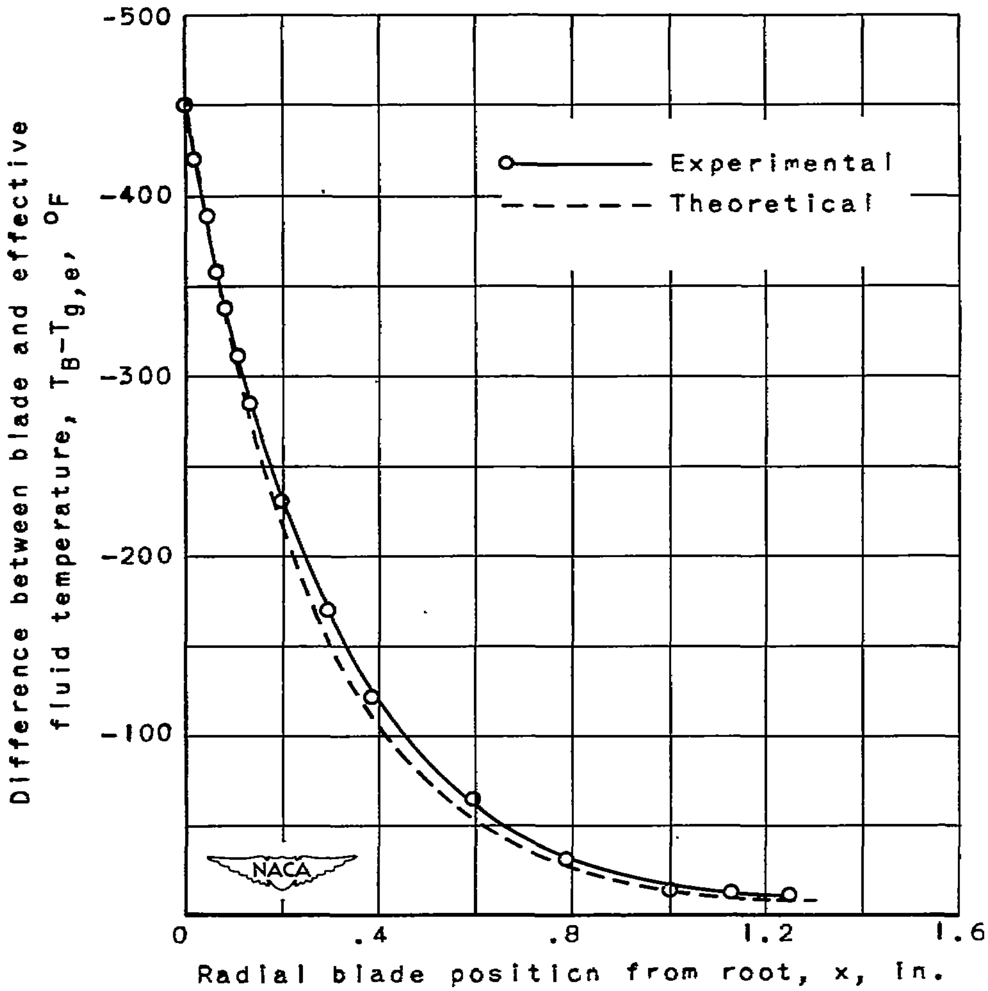
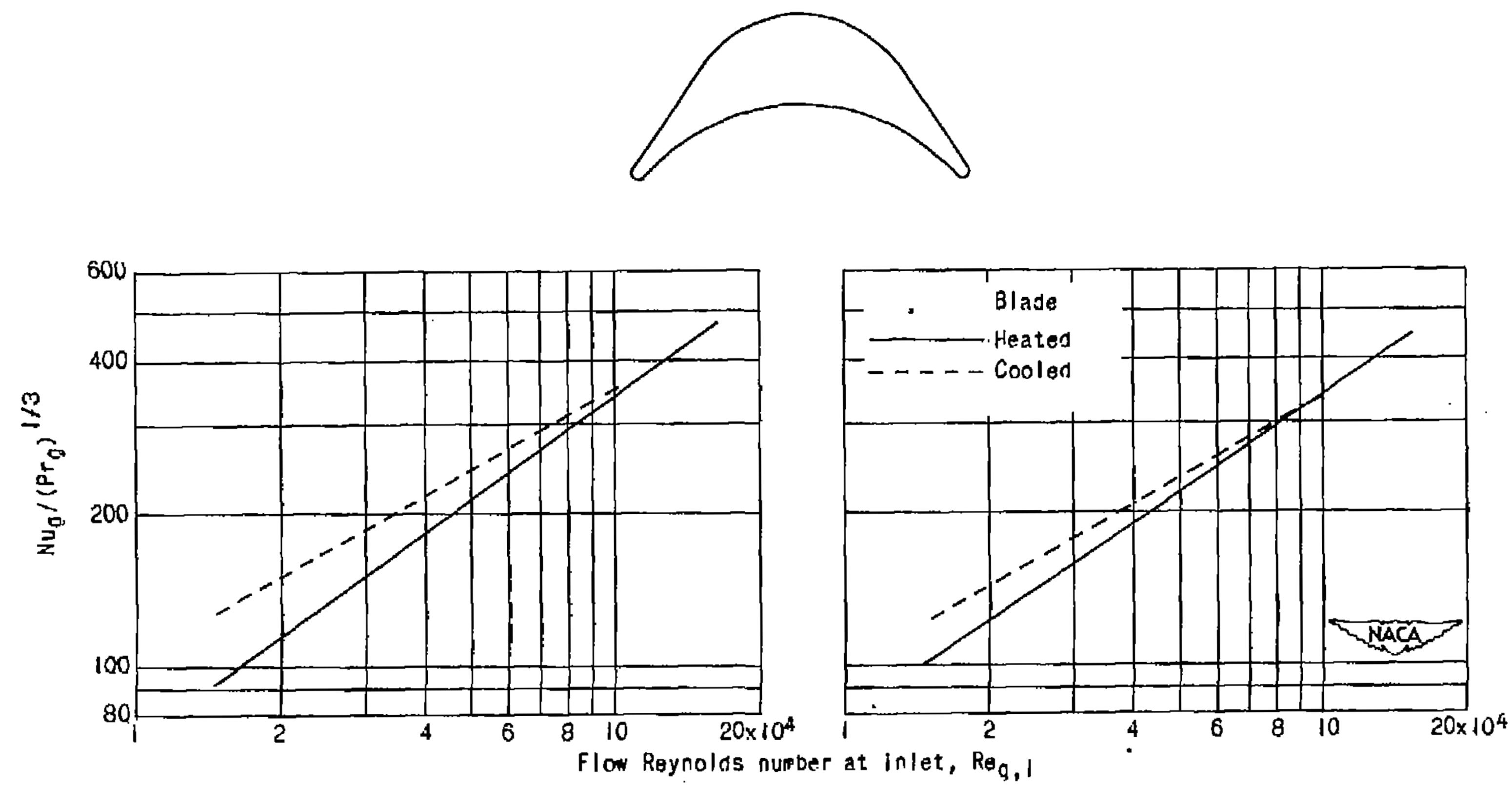


Figure 19. - Comparison of theoretical and experimental blade-temperature distribution at effective gas temperature of 1627° R.

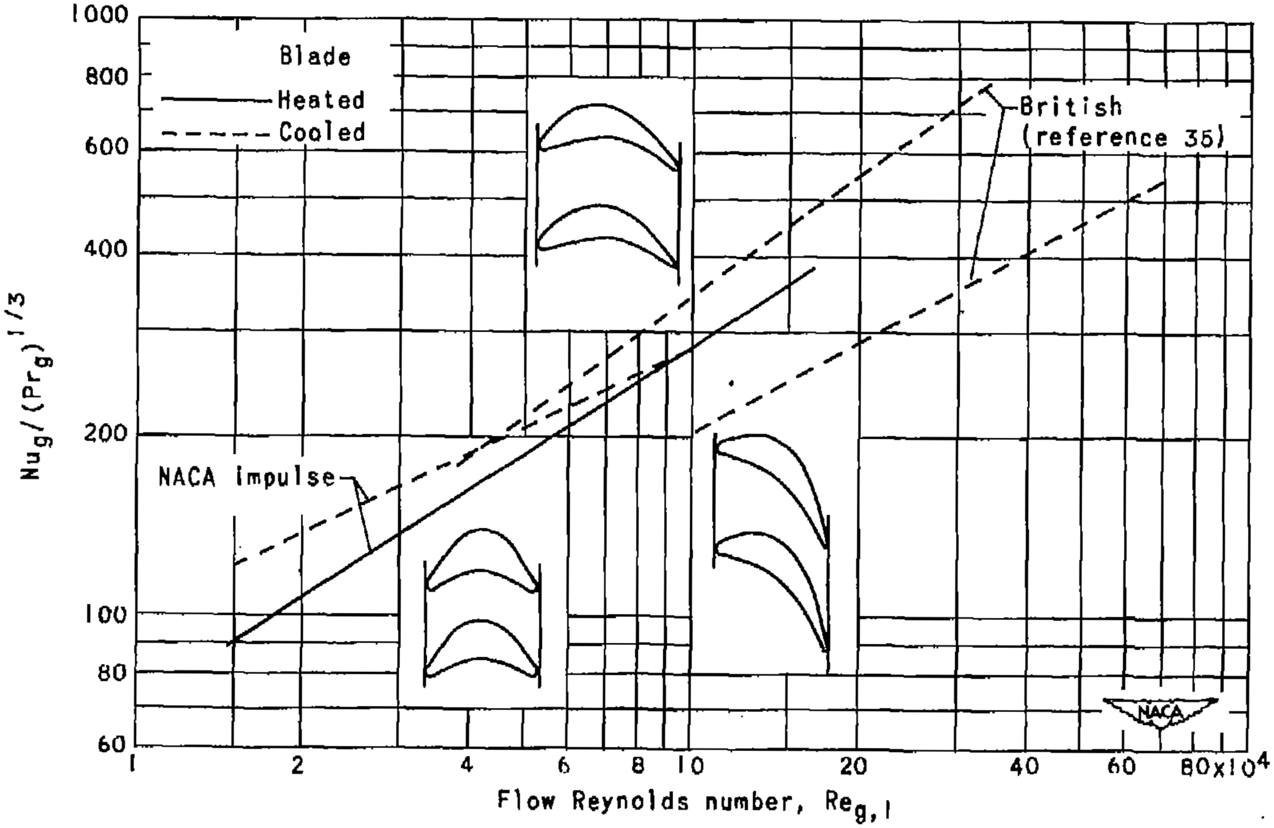


(a) Gas properties evaluated at film temperature; density at stream temperature.

(b) Gas properties and density evaluated at wall temperatures.

Figure 20. - Comparison of heat-transfer data for air flowing past NACA experimental impulse-turbine blades.

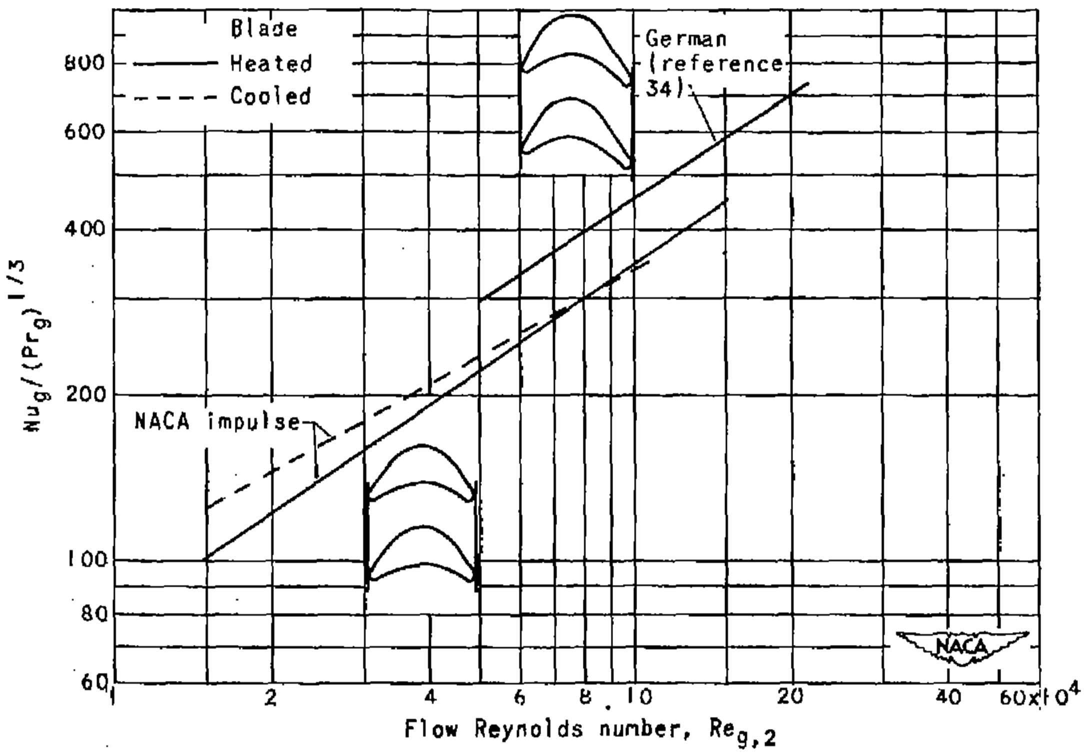
$$Nu_g = \frac{h_o Z_o/\pi}{k_g};\quad Pr_g = \frac{c_{p,g}\mu_g}{k_g};\quad Re_{g,1} = \frac{\rho_{g,1} W_{g,1} Z_o/\pi}{\mu_g}$$



(a) Gas properties evaluated at blade-wall temperature, density at mean of effective gas temperature and wall temperature, and mean of inlet and outlet pressure.

Figure 21. - Comparison of British, German, and NACA heat-transfer data.

$$Nu_g = \frac{H_0 l_0/\pi}{\mu_g};\quad Pr_g = \frac{c_{p,g}\mu_g}{k_g};\quad Re_{g,2} = \frac{\rho_g W_2 l_0/\pi}{\mu_g};\quad Re_{g,1} = \frac{\rho_g W_1 l_0/\pi}{\mu_g}$$



(b) Gas properties evaluated at blade-wall temperature; density at blade-wall temperature and inlet pressure.

Figure 21. - Concluded. Comparison of British, German, and NACA heat-transfer data.

$$Nu_g = \frac{H_o l_o/\pi}{\mu_g};\quad Pr_g = \frac{c_{p,g}\mu_g}{k_g};\quad Re_{g,2} = \frac{\rho_g W_2 l_o/\pi}{\mu_g};\quad Re_{g,1} = \frac{\rho_g W_1 l_o/\pi}{\mu_g}$$

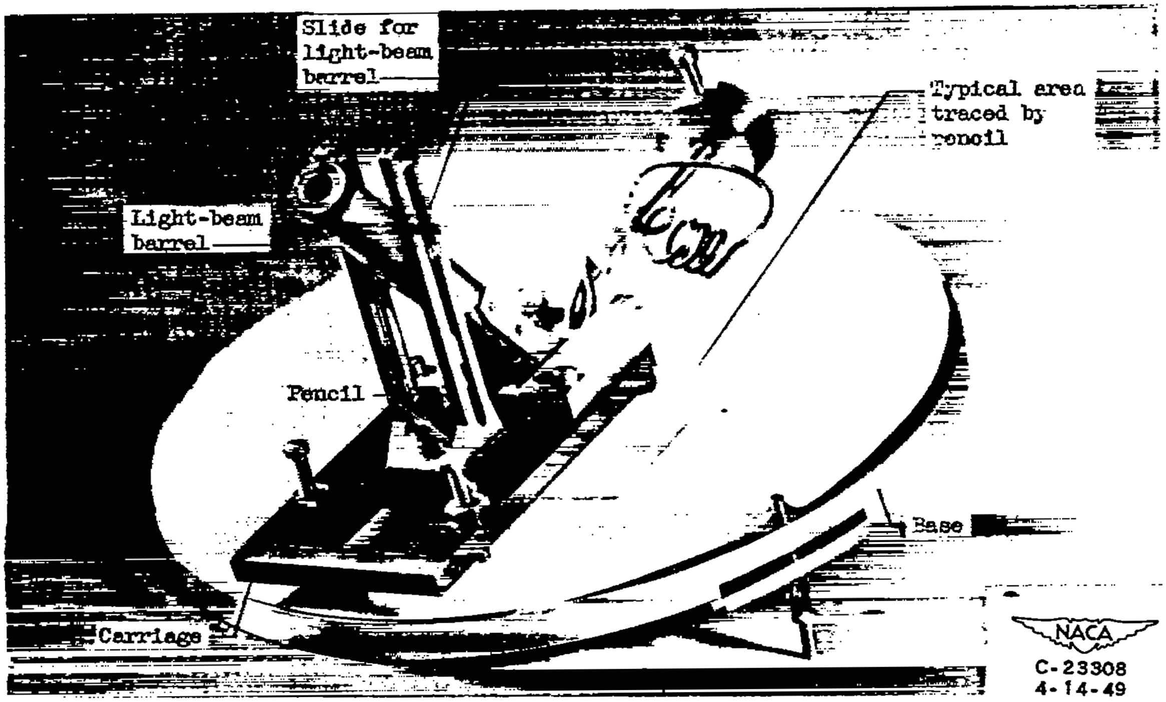


Figure 22. - Radiation integrator.

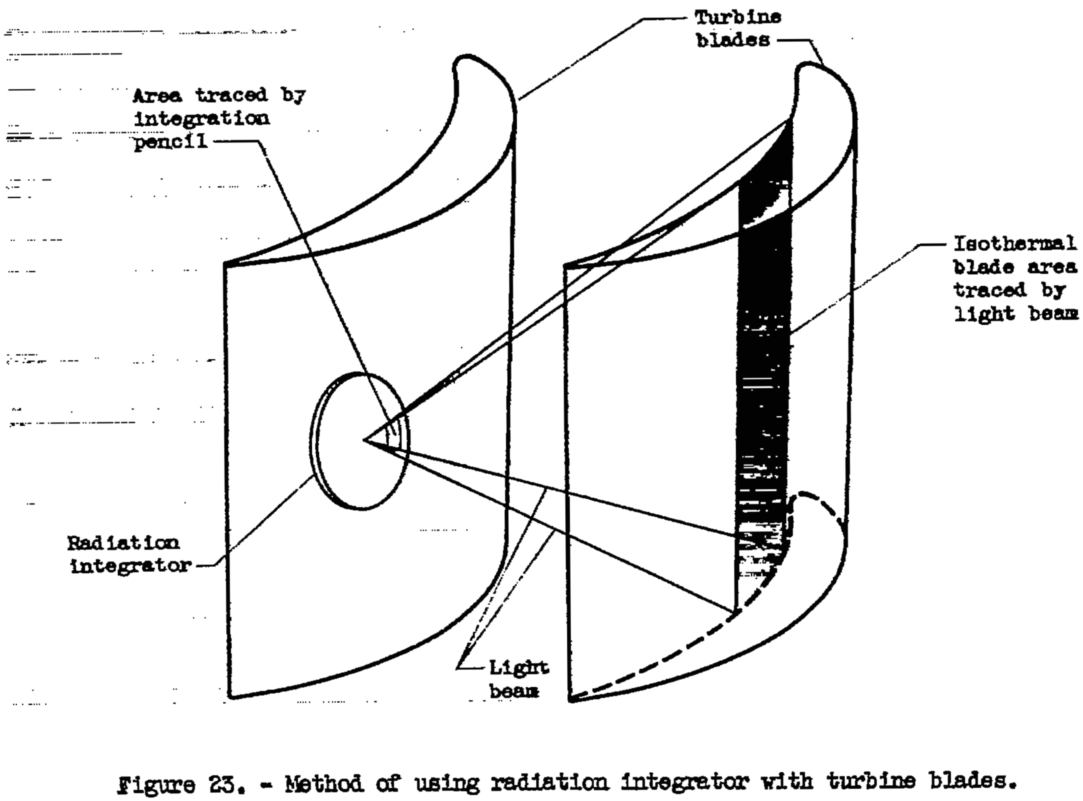


Figure 23. - Method of using radiation integrator with turbine blades.

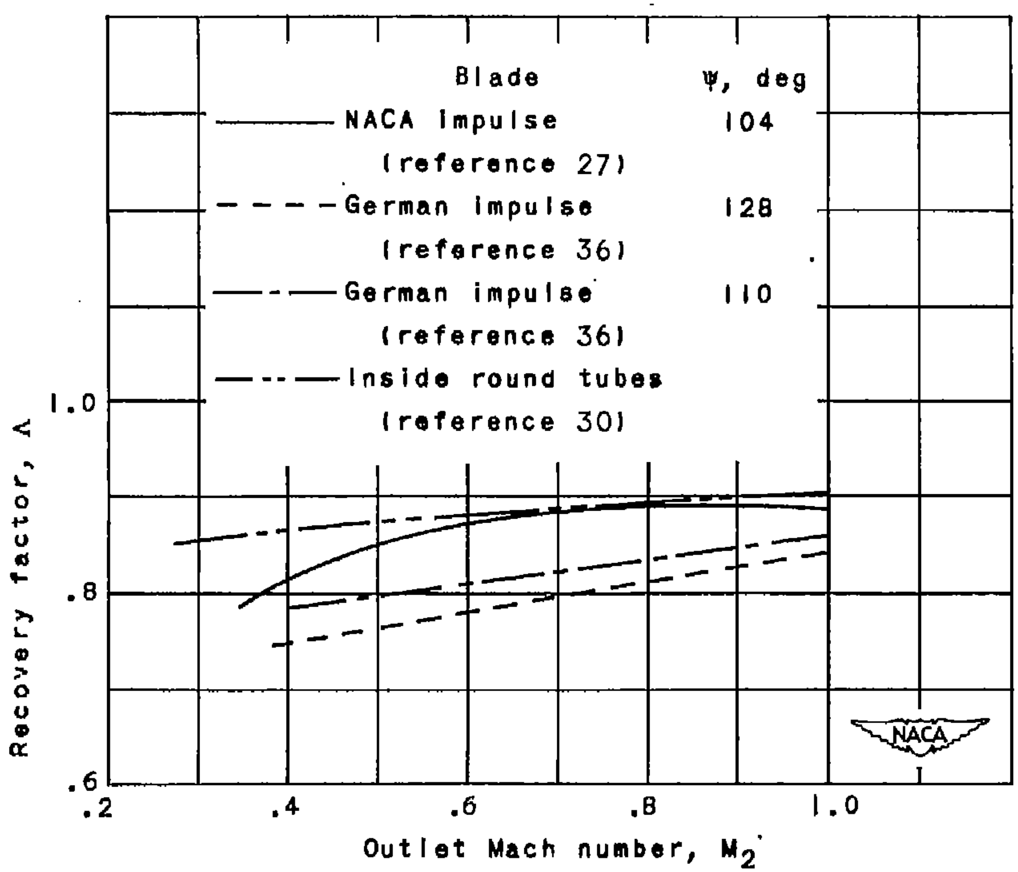
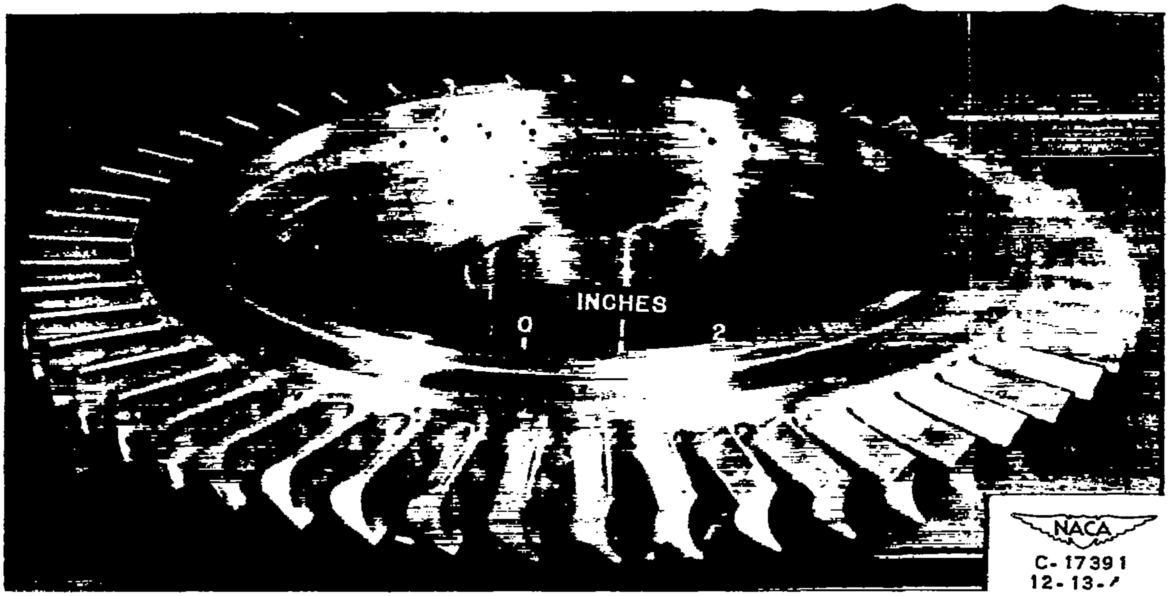
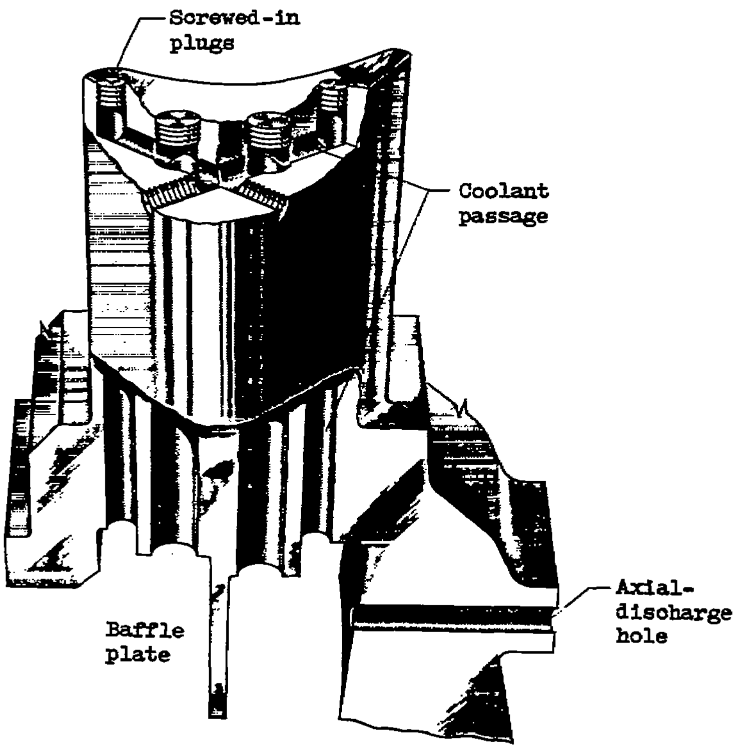


Figure 24. - Recovery factor for flow through round tubes and across impulse blades.



(a) Entire rotor.



(b) Section of turbine blade.

Figure 25. - Liquid-cooled aluminum turbine rotor.

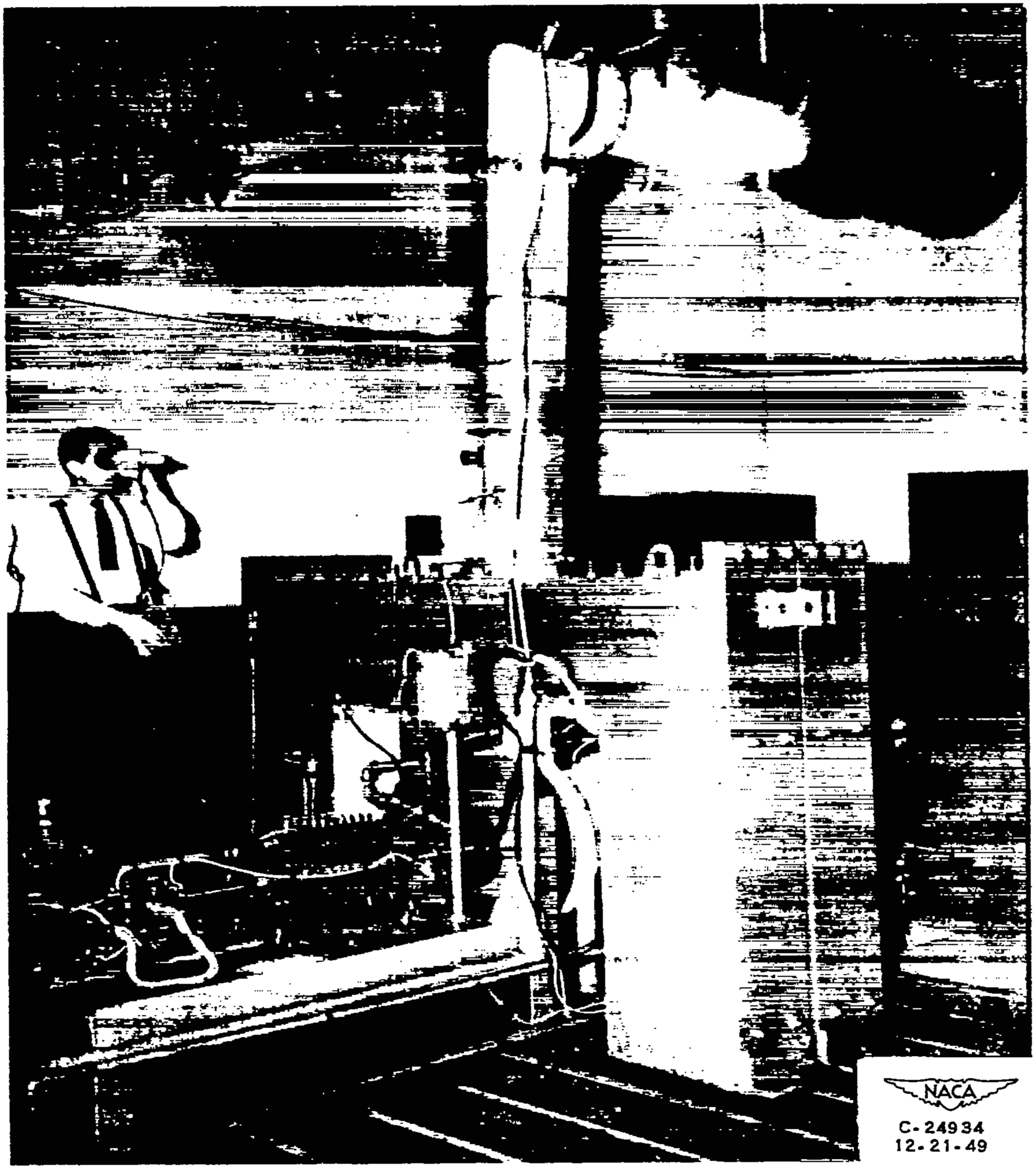


Figure 26. - Aluminum turbine in operation.

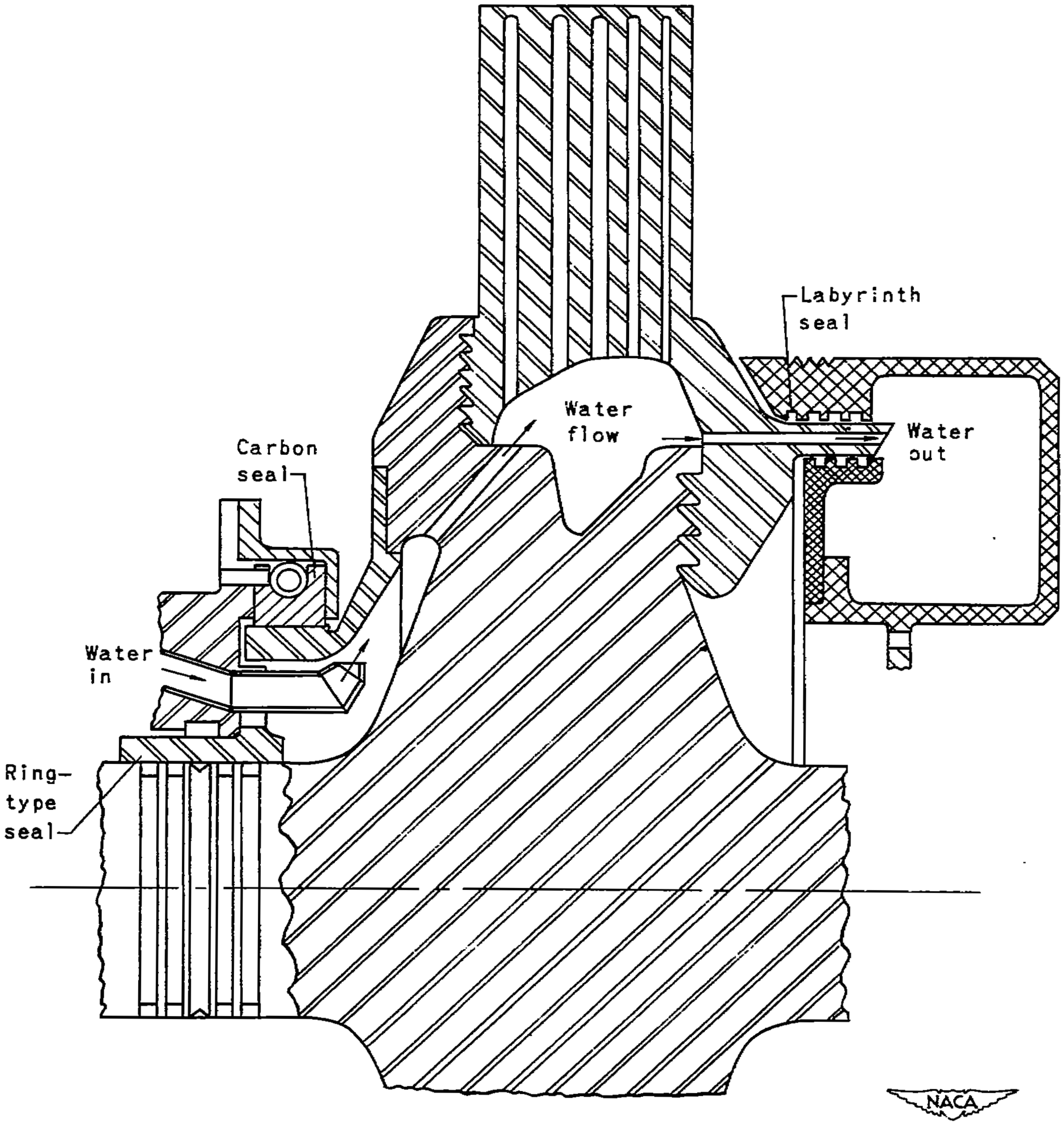


Figure 27. - Liquid-cooled steel turbine rig.

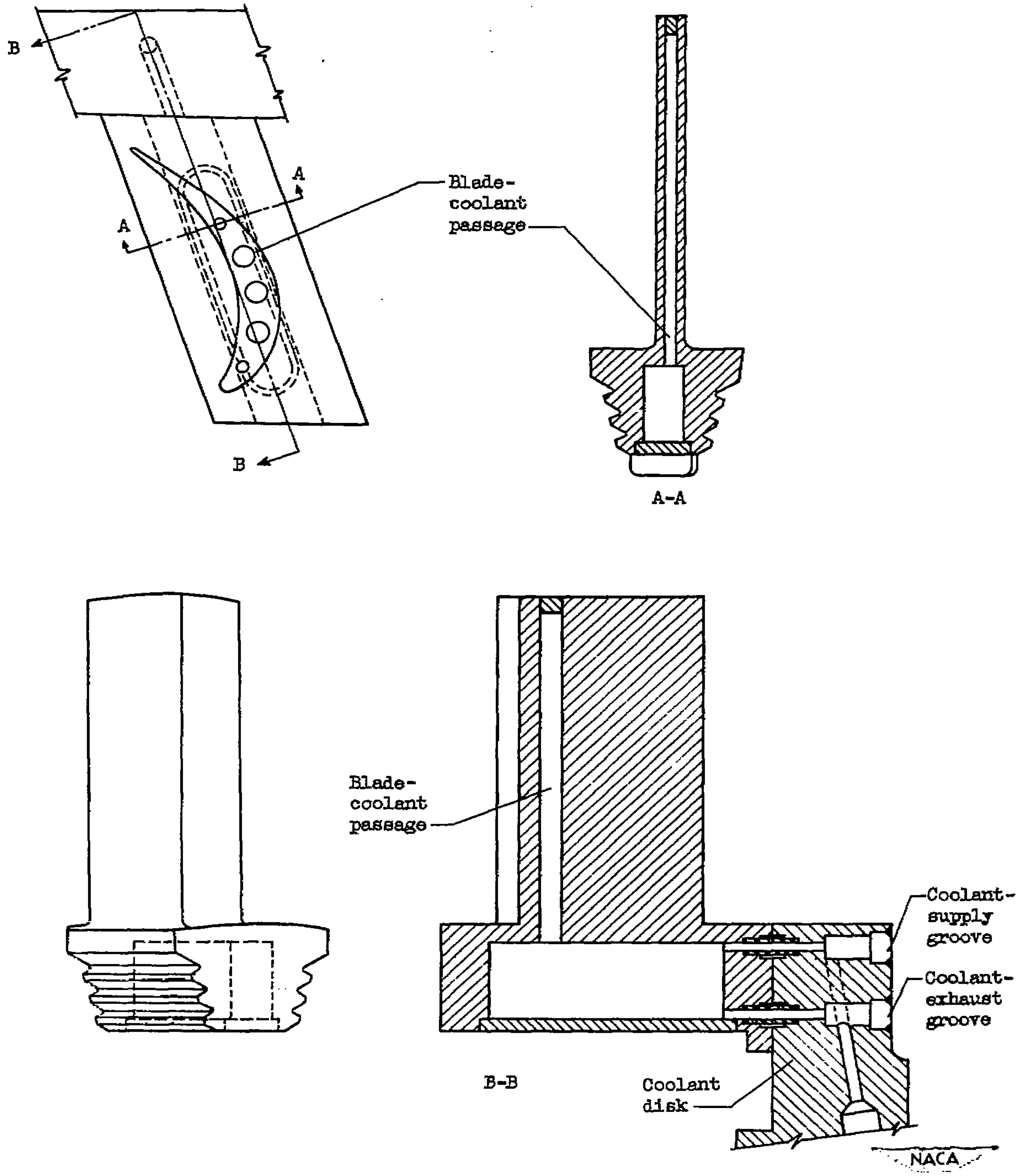


Figure 28. - Method of blade attachment in liquid-cooled steel turbine.